Final Design Report
Stator Insertion Machine Redesign

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Sponsored by:
Danfoss Turbocor

April 7th, 2011

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Executive summary

Stator insertion process redesign is sponsored by Danfoss Turbocor. The main goal of this project is to come up with a way to heat up an aluminum compressor housing quickly, efficiently and to the desired specification. The desired application for this design is to be directly implemented on the production floor.

Our sponsor representative Robert Parsons from Danfoss Turbocor had specific needs for the project. Reduce the overall size of the design, lower the final temperature at which desired expansion is achieved, keep the compressor housing clean and dry throughout the process, and achieve needed expansion in less than 15 minutes.

To reach the goals specified a number of tasks were performed. To get the project started first an idea was postulated, that only the desired diameter was of the concern. Using linear expansion as the fundamental law Experiment 1 was performed and verified the predicted expansion. Next Matlab was used to simulate the requirements for the heat source under varying conditions, including different ambient temperatures, various convection coefficients, and numerous losses associated with the process. At the end using the data a conclusion was achieved describing that a 5600 Watt electric heater could achieve the goal of heating up the compressor housing using forced convection to the desired temperature of 85 degrees Celsius in just under 8 minutes. There are still many steps required to complete the project, but there is enough data to support the fabrication and testing of a proof of concept and then a prototype to be used on the production floor.

Introduction

One of the most valuable commodities in production is time, and the less time you spend doing any one task, the more efficient your overall process becomes

Danfoss Turbocor is a company that designs and manufactures large industrial refrigerant compressors. Their compressor is the first to utilize magnetic bearings, a clever way to improve efficiency and decrease noise. They have determined that one of the first stages in their production line is too slow relative to the others. If this production point can improve its turn over time to approximately eight minutes it would be able to keep up with the rest of the facility and create a smoother, more efficient flow in their manufacturing line.

So the project Danfoss Turbocor brought to us is to improve this slow stage of their production. This stage is responsible for heating the compressor housing and then inserting a stator into it. The housing needs to be heated to allow for the thermal expansion of the metal to enlarge the housing and hole that the stator fits into, this allows for easy installation of the stator. The assembly then begins cooling which shrinks the housing around the stator to lock it in place.

The housing is cast from A356 T6 aluminum. Currently a large convection oven is used to heat four housings at a time. The heating time is roughly 45 minutes at a temperature of 300 °F and before the housing can continue through production it must cool for approximately one hour (Figure 1).

Figure 1: Current Heating Method

Problem Definition

The fundamental task of this project is to redesign the current method for heating the compressor housings. The primary goal of this project is to design a quicker method of heating the housing while scaling down the overall size of the unit. Ideally the system used will be able to heat the housing in less than fifteen minutes and keep the overall part temperature down which reduces cooling time and allows it to be handled easily by the operator.

Design Approach

The design approach involves defining the scope of the project and identifying the customer requirements. For the purpose of this project the first step is calculating thermal expansion. Once the linear expansion is calculated it must be verified through experiment. The next step is to calculate the heat transfer problem associated with the project. The analysis step is verifying the heat transfer calculations through proof of concept testing. Once completed, the actually design and assembly of the prototype can be completed. The design process can be seen through the flow chart (Figure 2).

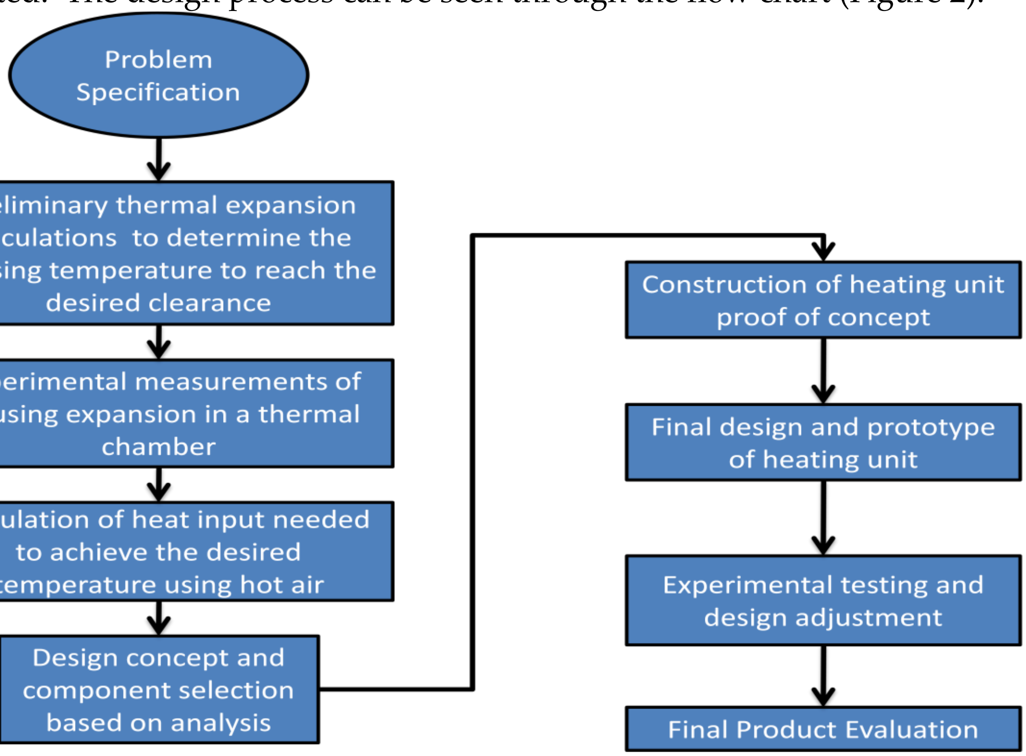


Figure 2: Design Approach

Concept Generation and Selection

For each of the following concepts the same six criteria are chosen to evaluate the designs based on their strong and weak characteristics. Performance includes efficiency, heating method, time required, and optimal operational temperature. Complexity deals with how simple the design would be to assemble and how easy it would be to replace expendable parts. Size is very important to the customer, that's why we included it as one of the main criteria. Durability considers what the expected life of the design is and how often the replaceable parts would need to be swapped out. Cost criteria was an obvious choice because like any business, saving money is always a plus, and finally user friendliness was chosen, since this is supposed to be installed on the production floor, so it should be easily operated by the necessary technicians.

Concept A – Convective Oven

Concept A uses forced convection to heat up the compressor housing. The insulated cabined would contain a powerful enough heater that would provide the necessary amount of heat input. Additionally a fan would re-circulate the air throughout the system. The air ducts would be incorporated into the cabinet to allow the air to be returned to the bottom and go through the cycle again. The compressor housing would simply slide horizontally on top of the cabined and be positioned in the desired location. Afterwards the insulated hood would be lowered on top to provide additional insulation and to help direct the air to the ducts that would re-circulate the air. Temperature probes could be inserted throughout the system to monitor the temperatures of the compressor housing.

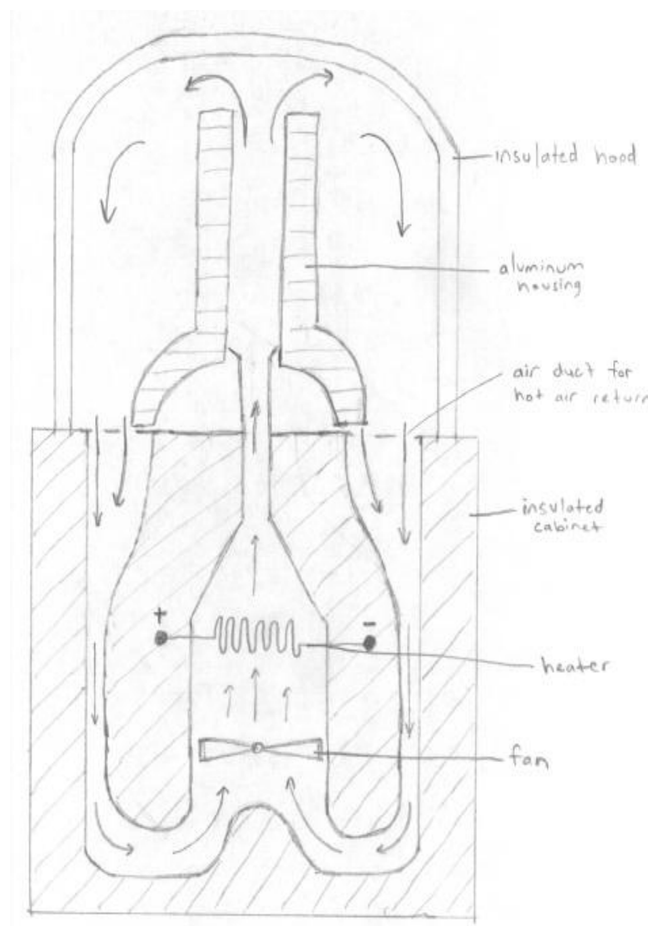


Figure 3: Concept A

Analysis:

The positive characteristics of this design are high performance and the easy implementation into the production cycle. The heater cores could easily be found and substituted for the desired amount of heat input. The fan speed could also be regulated to control the forced convection. The fan would be appropriately chosen for the desired application, because of the high operating temperatures. The idea behind this concept is very simple because there are only two components to it, the heater and the fan. The compressor housing itself is used to channel the air and then the hood to return the air. The overall size of the design is comparably small because all the heater components are contained below in the insulated cabinet and because the stator insertion is a next process not incorporated in this design. The design is of average durability because the heater coils are subject to high operating temperatures, and will be the first thing to fail. Also the fan should be appropriately chosen not only to be able to provide the necessary air flow but also to be able to operate in the high temperatures required. The cost of the design is a little higher than some of the other concepts because of the requirements of the specific heater, fan and the required insulation. This is a very operator friendly concept because it only requires the housing to be moved into an appropriate position one time. Once the housing is located to the stator insertion mechanism no further manual input would be required.

Concept B – Oil Bath

The idea behind this concept is to use hot fluid to heat up a part. Since water cannot be heated above 212 degrees Fahrenheit it is not a good option for use as the heating fluid. Mineral oil on the other hand can be heated to about 500 degrees Fahrenheit without boiling, making it a good choice as the heating fluid. Now in order to heat the oil, an inductively heated steel tank is used, the steel tank would then conduct the heat into the fluid. As shown in Figure B the steel tank is designed for this specific housing to fit in it and around it by having a cylindrical extension in the middle to help heat the oil. The oil bath must be brought up to temperature before the housing is inserted because the induction heater can damage the aluminum housing when in operation. So the oil tank is inductively heated to about 500 degrees Fahrenheit and then the housing is inserted into the hot fluid where conduction heat transfer takes place. This method allows for a uniform heating of the part which eliminates residual stresses in the material.

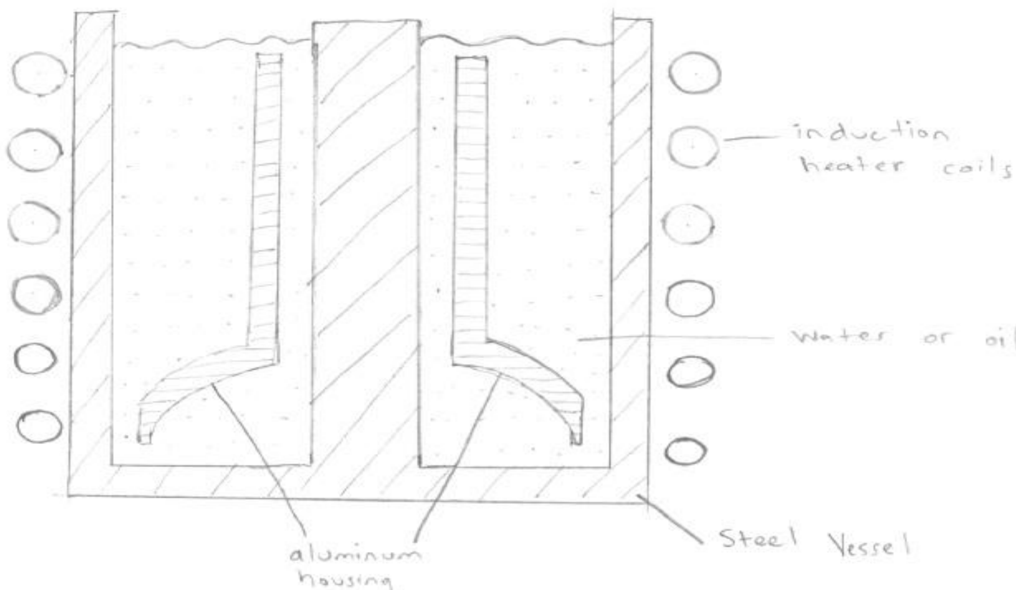


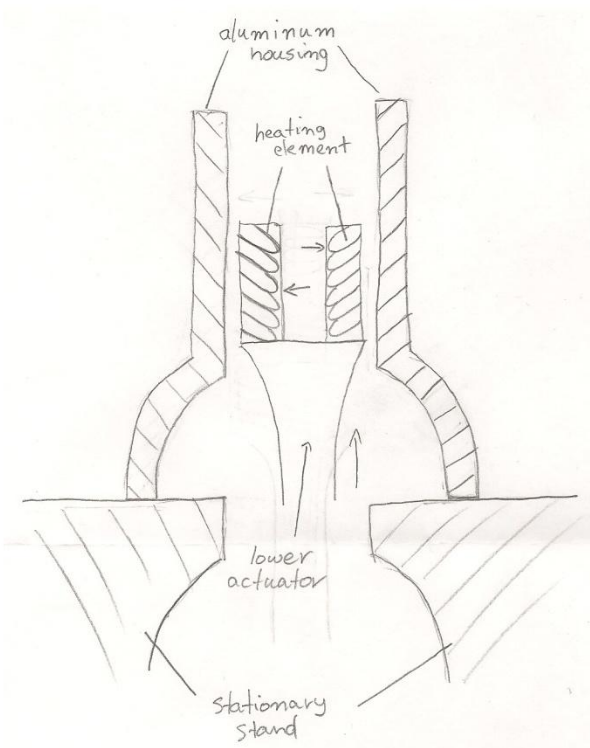
Figure 4: Concept B

Analysis:

This method of heating works well although the time needed to heat the part may become an issue. In this case time is an issue so the faster the part can be heated up the better. Adding a circulating pump to allow convection within the fluid would be an improvement to this design by increasing performance through time needed to heat up the part, but may slightly decrease durability while increasing complexity. This concept utilizes commonly available materials and equipment making it a rather simple design. The steel tank would be the most complex since it is specially designed for this part. This unit would be rather large since the part needs to fit inside it along with enough working fluid to effectively transfer enough heat. This is a problem for this facility due to limited floor space. With no moving parts except a possible circulating pump, this unit can basically last forever. If failure were to occur it would be within the induction heater itself. Another consideration is the replacement of oil as a small percentage of it will be lost throughout operation. Initial costs are relatively low since the parts and materials used are readily available and inexpensive. And maintenance costs will be low as well since the durability is so high but oil may need to be kept on hand for replacement and or topping off. This method is not very user friendly, it involves placing and removing hot parts from very hot oil, operator safety may be an issue.

Concept C – Internal Resistive Heating

This concept utilizes small size, relative low cost and ease of implementation into the existing production cycle. The concept consists of 2 pneumatic actuators. As the compressor housing comes into position on the production table, the lower actuator inserts and expands the heating element into the housing to provide conductive heating throughout the required area. After the required temperature has been reached, the lower actuator would lower and the same time the upper actuator would insert the preassembled stator into the housing. Afterwards the upper actuator would retract and grip another stator, while the assembled housing slides down the line for cooling and continuing assembling.



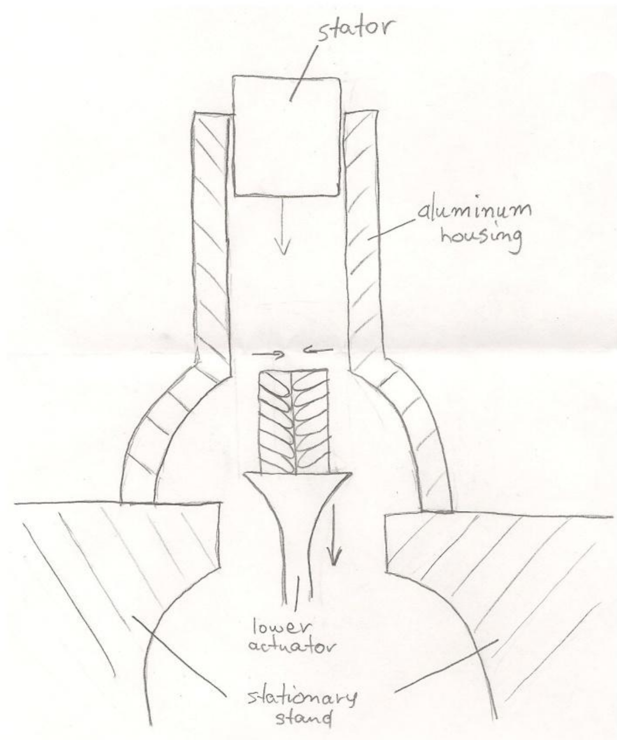


Figure 5: Concept C

Analysis:

Compared to the other concepts this design would have average performance, since the heating coils would heat up quickly but the transfer of this heat into the housing is compromised by the fact that the element does not touch the housing surface evenly throughout the required area. Also because only stator area of the housing is heated, there will stresses associated with uneven temperature distribution. This system would be relatively complex since there are a lot of moving parts, both of the actuators and the expansion of the heating element itself. Also because of the actuators precise positioning would be required. The overall size of the design would be a lot smaller than the other concepts because there is no need for any tanks or hoods and the heating coils are only as big as the compressor housing. Durability would suffer because again there are a lot of moving parts which will need to be properly maintained. Also there is hardly any way of predicting when the copper coils of the heating element will go bad, thus replacement ones will need to be on hand for substitution. The cost will go up because of low durability. Also constant maintenance will be required thus making it even more expensive to run. Replacement parts will need to be kept on hand which increases cost because even though the parts are paid for, they are not being used. After the initial installation setup, the operations would actually be very easy since the only actions required are the positioning of the housing, and initializing the start of the cycle.

Concept D – Internal Induction Heating

Another concept in heating the inside of the stator housing is to use internal inductive conduction. This just means that the heat will be dispersed by contact of a metal that will be heated up by an electric current that is caused to flow through the metal to the internal walls of the cylinder. This copper will heat up by having current going up through it from a battery source. The process of heating the housing is, first to have the housing elevated over top of the copper cylinder. The copper cylinder will then be lifted up into the housing (lip side). Once fully in the housing, the copper cylinder will split open in four symmetrical quarter circles until each section is in contact with the inner wall of the housing. At that point, the current will go through the copper causing it to heat up to the temperature determined through analysis. This will cause the inner housing walls to expand to the desired dimension. Finally, the copper quarter circles will collapse back together and drop down out of the housing. Then the stator will be pressed into the heated and expanded housing and cooled to shrink fit stator in the housing.

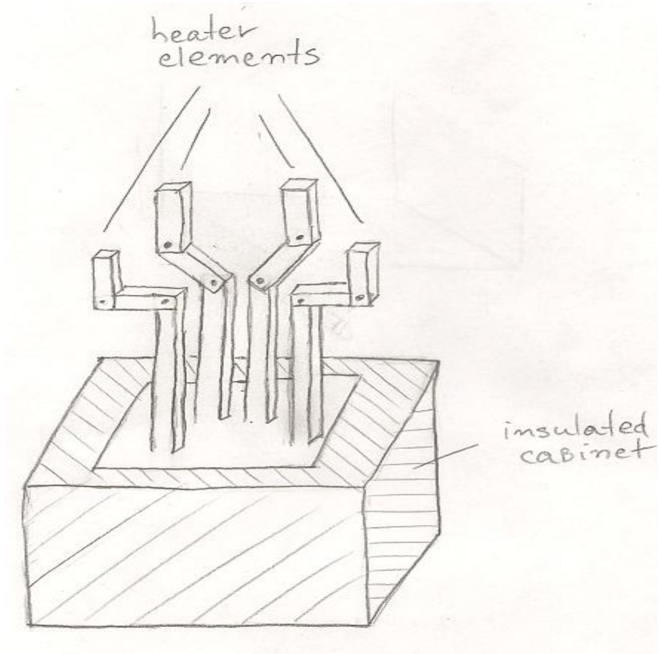


Figure 6: Concept D

Analysis:

The copper can reach a significantly high temperature with no issue. And will be able to disperse the heat throughout the inner wall of the housing evenly. This process should take no more than 15 minutes to heat and expand the inner walls, depending upon the amount of energy exuded. There are a lot of parts and mechanisms that are in this concept, causing it to be very complex. The most detailed area is the actual design of the shape of the soon-to-be heated copper, because the geometry of it has to be in a way that when it is expanded, it can be in contact with at least 90% of the inner wall of the housing. The linkages that are going to hold the copper and lift it will either be in copper or preferably steel so that the time for the copper to heat up will be shorter. The dimensions of the machine will not be big at all. It will be longer in height if anything because of the raises and lowering of the copper into the housing. With the mechanical arms being steel and the heat exchanger being copper, it makes the machine very durable. Both metals have high density, toughness, and fracture toughness. If any failure it will be in the joints of the mechanical arms. The economic value for this concept is not too expensive because steel, copper, joints, and a battery source are pretty easy to come by because they are heavily used in the industry. The most expensive part is the motor that will lift the mechanical arms and open them. The whole machine will be run by a controller that will direct it what to do; this makes it very safe and easy to use in any work place. The only part that may be done by human force is the removal of the completed shrink fitted stator in housing to the cooling area.

Concept Selection

The following decision matrix was constructed to aid in the making the decision. The weights are assigned to each criteria based on how important they are to the customer. Performance and size make up 50% of the overall design because those were the two most emphasized design characteristics. The rating is on a 1 to 5 scale, one being the lowest and five being the highest.

Table 1: Decision Matrix

Concept Selection									
		Convective Oven		**Oil bath**		**Internal Resistive**		**Internal Induction**	
Selection Criteria	Weight	Rating	Weight Score	Rating	Weight Score	Rating	Weight Score	Rating	Weight Score
Performance	30%	4	1.2	5	1.5	3	0.9	3	0.9
Complexity	20%	4	0.8	3	0.6	2	0.4	2	0.4
Size	20%	3	0.6	2	0.4	4	0.8	3	0.6
Durability	15%	3	0.45	4	0.6	1	0.15	2	0.3
Cost	10%	2	0.2	2	0.2	3	0.3	3	0.3
User Friendly	5%	3	0.15	1	0.05	3	0.15	3	0.15
Total		3.4		3.35		2.7		2.65	
Ranking		1		2		3		4	

Based on the decision matrix it was decided that Concept A – Convective Oven was chosen as our final design. Even though it has almost the same ranking as the Concept B, after presenting the choices to the customer, he eliminated the oil bath because the compressor housing must remain dry. Another idea that was not analyzed in full detail was cooling the stator using liquid nitrogen. There was concern that cooling the stator could potentially damage the stator windings. Our sponsor verified this concern and rejected the idea.

System Analysis

Linear Expansion

The stator utilizes an interference fit with the housing. The fit is at maximum material condition between the stator and the housing. The required clearance for this type of fit is based on the clearance fit section of the machinist handbook. The insertion process utilizes a basic shaft into hole free fit, the appropriate clearance for this is 60 microns. The temperature needed to reach the desired clearance was calculated using the thermal expansion relationship (Equation 1). The dimension of interest was the internal diameter of the housing. The linear expansion equation is valid because we are not concerned with how the rest of the housing expands, only the inner diameter.

$$\boldsymbol{\Delta L = L_o * \alpha * \Delta T} \qquad \textbf{(1)}$$

Based on the supplied drawings the smallest diameter of the housing is 0.1715m, the maximum stator dimension is 0.1716m. The coefficient for thermal expansion for aluminum A356 is 0.00023 m/m°C. Ambient temperature is considered 25°C. Using this information the clearance between the two parts was calculated based on the final temperature of the housing (Figure 7). From the graph the final temperature of the housing is approximately 85 °C (185°F). This final temperature is based on a 25 °C starting temperature. This means that the necessary change in temperature to achieve 60 microns clearance is 60 °C.

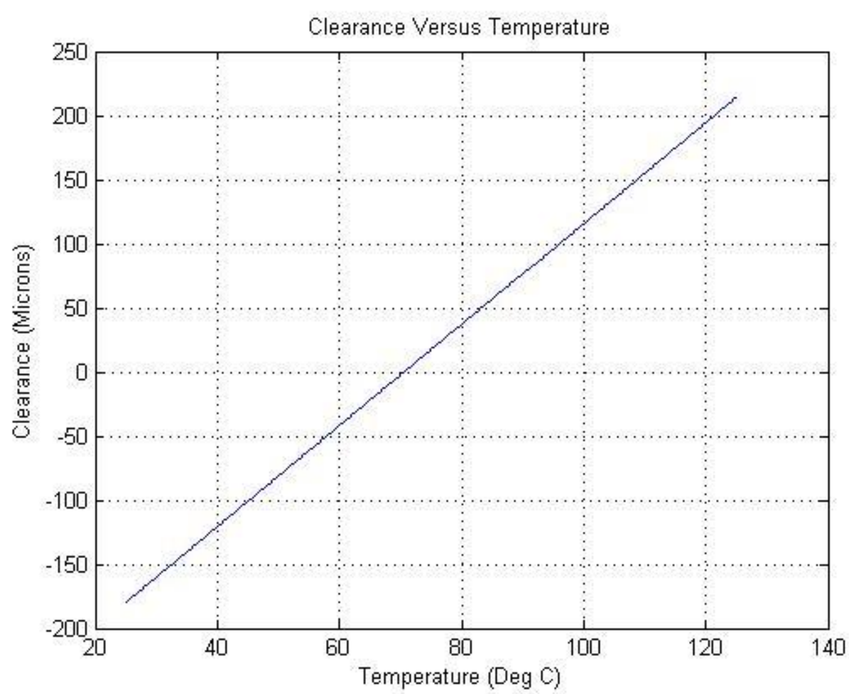


Figure 7: Linear Expansion versus Temperature

Internal Convection Coefficient

The internal heat convection coefficient is the primary mechanism of heat transfer to the housing. The nozzle attached to the end of the heater will promote turbulence and increase heat transfer. For our analysis the volumetric flow rate was varied until turbulent flow was achieved. Many of the parameters needed to calculate the convection coefficient are based on temperature of the medium. Our heating process is a transient problem in which the air temperature is constantly changing. In order to establish a temperature to find these properties at we needed to know the average air temperature. A MatLab program was created so that we could iterate until the average air temperature could be found. The initial properties were guessed. The program then finds the air temperature. The properties are then found for this new temperature. This process is completed until the temperature converges.

The first step was calculating the density of the air in the system. Air was treated as an ideal gas for our application because the reduced temperature was greater than 2, and the operating pressure was substantially lower than the critical pressure. The density was calculated using the ideal gas law (Equation 2). The next step was to calculate the Reynolds number so the proper Nusselt number correlation could be used (Equation 3).

$$\rho = \frac{P}{RT} \tag{2}$$

$$Re = \frac{\rho V D_h}{\mu} \tag{3}$$

For our system we simplified the housing geometry to a simple cylinder and treated the flow as flow within a pipe. The Reynolds number was 50,766. For turbulent flow in a tube the Dittus-Boelter equation was utilized to calculate the Nusselt number (Equation 4). Finally the convection coefficient was calculated for our system (Equation 5).

$$Nu = 0.023 Re^{0.8} Pr^{n} \tag{4}$$

$$h_i = \frac{Nuk}{D_h} \tag{5}$$

A detailed table of all the key property values can be found in Appendix II. The internal convection coefficient for our system based on a nozzle diameter of 10cm and a volumetric flow rate of 250 CFM was 15.9 W/m²K.

External Convection Coefficient

Natural convection is the mechanism of heat transfer from the exterior of our hood to the surrounding environment. Calculating the natural convection coefficient was a similar process to the forced convection coefficient. Many of the key properties needed are based on the film temperature, or the average between the surface and ambient temperatures. This again was an iterative process. Using the resistance network in the following section the properties were guessed. The total heat transfer was found and the surface temperature was solved for. This new temperature was then used to find the new film temperature. This process was repeated until the surface temperature converged. One of the key parameters in natural convection is the Graschof number (Equation 6). The Graschof number is a ratio of the buoyancy to viscous force acting on the fluid. The next step was calculated the Rayleigh number (Equation 7). Using the vertical plate correlation the Nusselt number was calculated (Equation 8). Finally the convection coefficient was calculated (Equation 9).

$$Gr = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2} \tag{6}$$

$$Ra = GrPr \tag{7}$$

$$Nu = 0.59Ra^{\frac{1}{4}} \tag{8}$$

$$h_o = \frac{Nuk}{L_c} \tag{9}$$

A detailed table of all the key property values used for analysis can be found in Appendix II. Once the Nusselt number is calculated the natural convection coefficient was calculated. For our system the natural convection coefficient was 4.89 W/m²K.

Thermal Resistance Network

In order to calculate the heat loss from our system the thermal resistance approach was utilized. The total thermal resistance was the sum of the forced convection inside, the thermal conductivity of our hood material combined with the insulation, and the external natural convection (Figure 8).

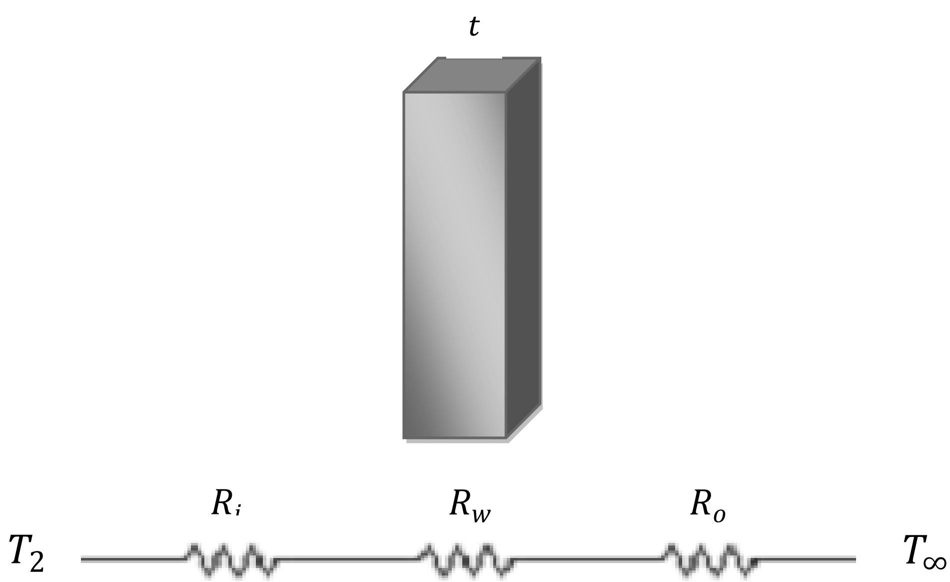


Figure 8: Thermal Resistance Network

The primary variables in the thermal resistance network are the hood material properties. For our hood material our first choice was sheet metal with a layer of insulation. The thermal conductivity of plate is 42 W/mK. The thermal conductivity of the insulation is 0.16 W/mK. For our system the wall thickness was 10cm. The material and thickness are system variables that can be modified as needed during heat transfer analysis. In the figure the inside resistance is due to convection from the internal air to the hood surface (Equation 10). The wall resistance is the thermal resistance of the hood material (Equation 11). The outside resistance is due to natural convection from the hood to the ambient conditions (Equation 12).

$$\boldsymbol{R_i = \frac{1}{h_i A_{heated}}} \tag{10}$$

$$\boldsymbol{R_w = \frac{t}{k_{metal} A_{heated}} + \frac{t}{k_{insulation} A_{heated}}} \tag{11}$$

$$\boldsymbol{R_o = \frac{1}{h_o A_{heated}}} \tag{12}$$

All three resistances utilize the heated area. A detailed table of all the key property values can be found in Appendix II. For our system this is the hood surface area which is 3.29 m^2. The final calculation of our total thermal resistance was 0.1535 K/W.

Control Volume Analysis

The next phase in the analysis was to calculate the size of the heater we would need for our convection system. For analysis we considered the entire heating unit as a closed system. The main control volume will encompass the entire unit with a second control volume inside of the system (Figure 9).

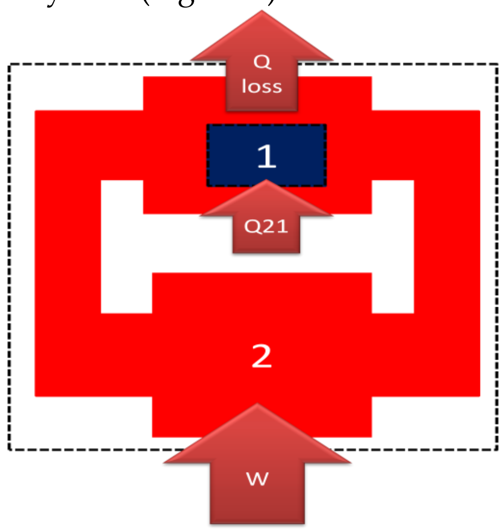


Figure 9: Control Volume

The only heat input into system one is the convective heat transfer from system two. System two loses heat to system one and through losses through the system. System two also receives power input from the heater. In order to complete the analysis the entire system was decomposed into a coupled system of two control volumes (Figure 10).

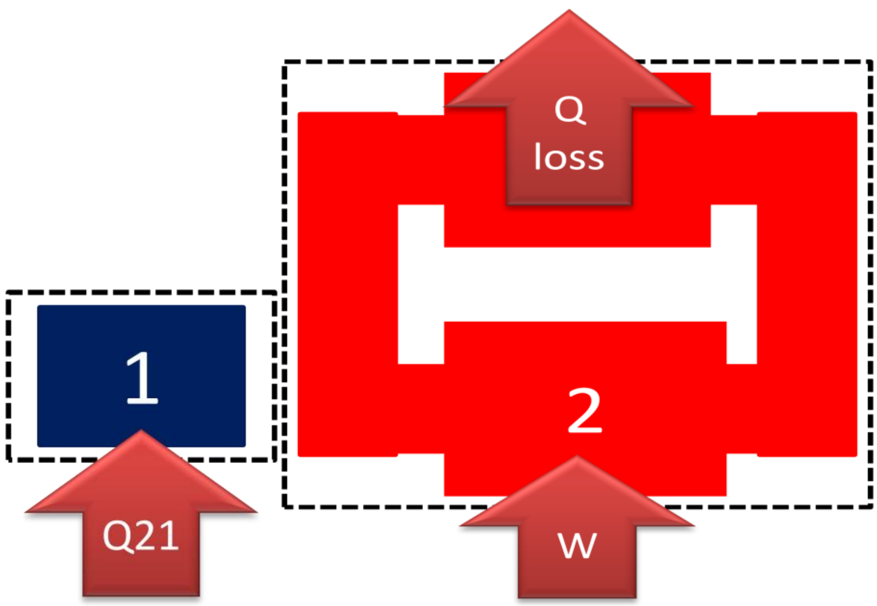


Figure 10: Control Volumes

Using the first law of Thermodynamics, equations were derived for the change in internal energy of the systems (Equation 13&14).

$$\boldsymbol{m_1 C_1 \frac{dT_1}{dt} = \dot{Q}_{21}} \tag{13}$$

$$\boldsymbol{m_2 C_v \frac{dT_2}{dt} = \dot{W} - \dot{Q}_{12} - \dot{Q}_{loss}} \tag{14}$$

The heat term from system two into system one is convective heat transfer from the air into the housing (Equation 15). The work term is the power rating of our heater. The heat loss term is based on a thermal resistance network from the inside of the system to the exterior environment (Equation 16).

$$\boldsymbol{\dot{Q}_{12} = h_i A_1 (T_2 - T_1)} \tag{15}$$

$$\boldsymbol{\dot{Q}_{loss} = \frac{(T_2 - T_\infty)}{R_{tot}}} \tag{16}$$

Simplifying the previous equations we end up with a coupled system of ordinary differential equations (Equation 17&18).

$$\boldsymbol{\frac{dT_1}{dt} = \frac{h_i A_1 (T_2 - T_1)}{m_1 C_1}} \tag{17}$$

$$\boldsymbol{\frac{dT_2}{dt} = \frac{\dot{W}}{m_2 C_v} - \frac{h_i A_1 (T_2 - T_1)}{m_2 C_v} - \frac{(T_2 - T_\infty)}{C_v R_{tot}}} \tag{18}$$

Using MatLab a graph relating the temperature of both systems was generated (Figure 11). The MatLab code was created so that the heater value could be varied in order to find a sufficient heater in order to meet our heating time design requirements. Modulating the heater led us to the conclusion that a 5600W heater would allow us to achieve our goal.

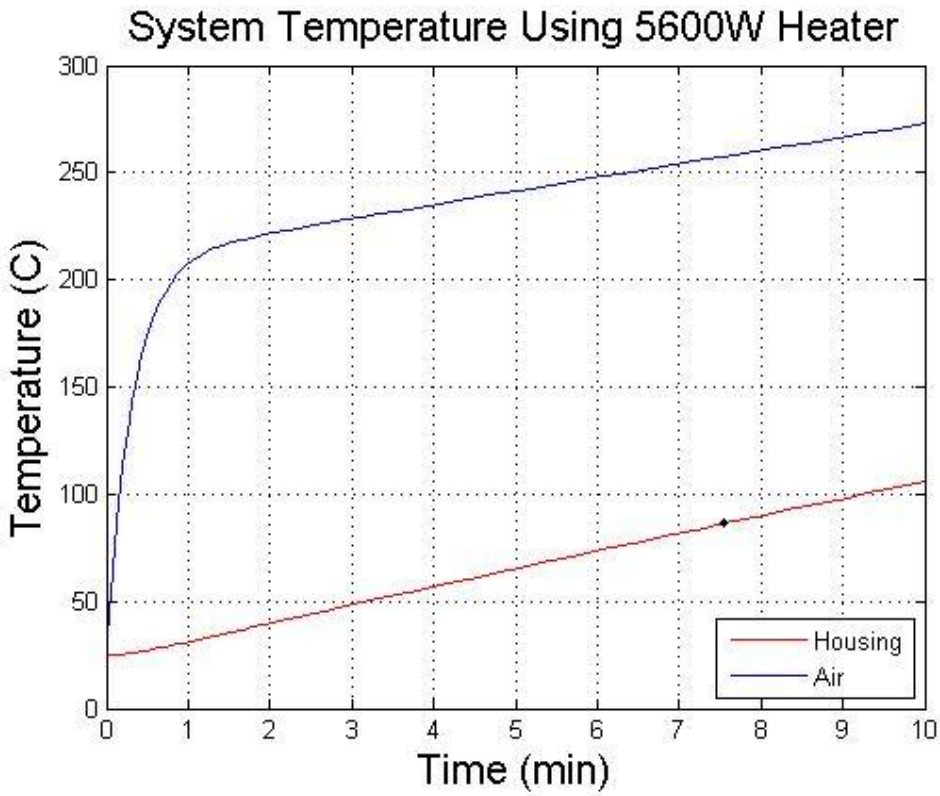


Figure 11: System Temperature versus Time

We should be able to reach the desired temperature of 85°C in less than 15 minutes. According to the plot we will achieve our goal in nearly 8.5 minutes. Our MatLab simulation allows for this adjustment by changing the necessary values. A detailed table of all the key property values can be found in Appendix II.

Experimental Data

Experiment 1: Verifying Linear Expansion

Experiment 1 was performed on site, at Turbocor to verify the linear expansion of the compressor housing. The housing was heated to more than 110 degrees Celsius and then the inner diameter and housing temperature measurement were taken with the bore gauge and thermocouples at specific time intervals as the housing was cooling down in ambient air. We were unable at measure the exact diameter where the stator would be located. To compensate a different diameter was measured. The data was then non-dimensionalized to compare the data to the expected trends. Once the experiment was completed we verified that the linear expansion relationship was valid (FIG 10). Ideally we would perform this experiment numerous times while testing many different housing locations. Unfortunately due to the nature of the manufacturing process at Turbocor, our oven time was limited. We feel our data does still verify that the critical final temperature of 85 °C will work for the purposes of this design. This final temperature is based on an initial temperature of 25 °C. Because the expansion formula is linear the actual change in temperature from needed is 60 °C.

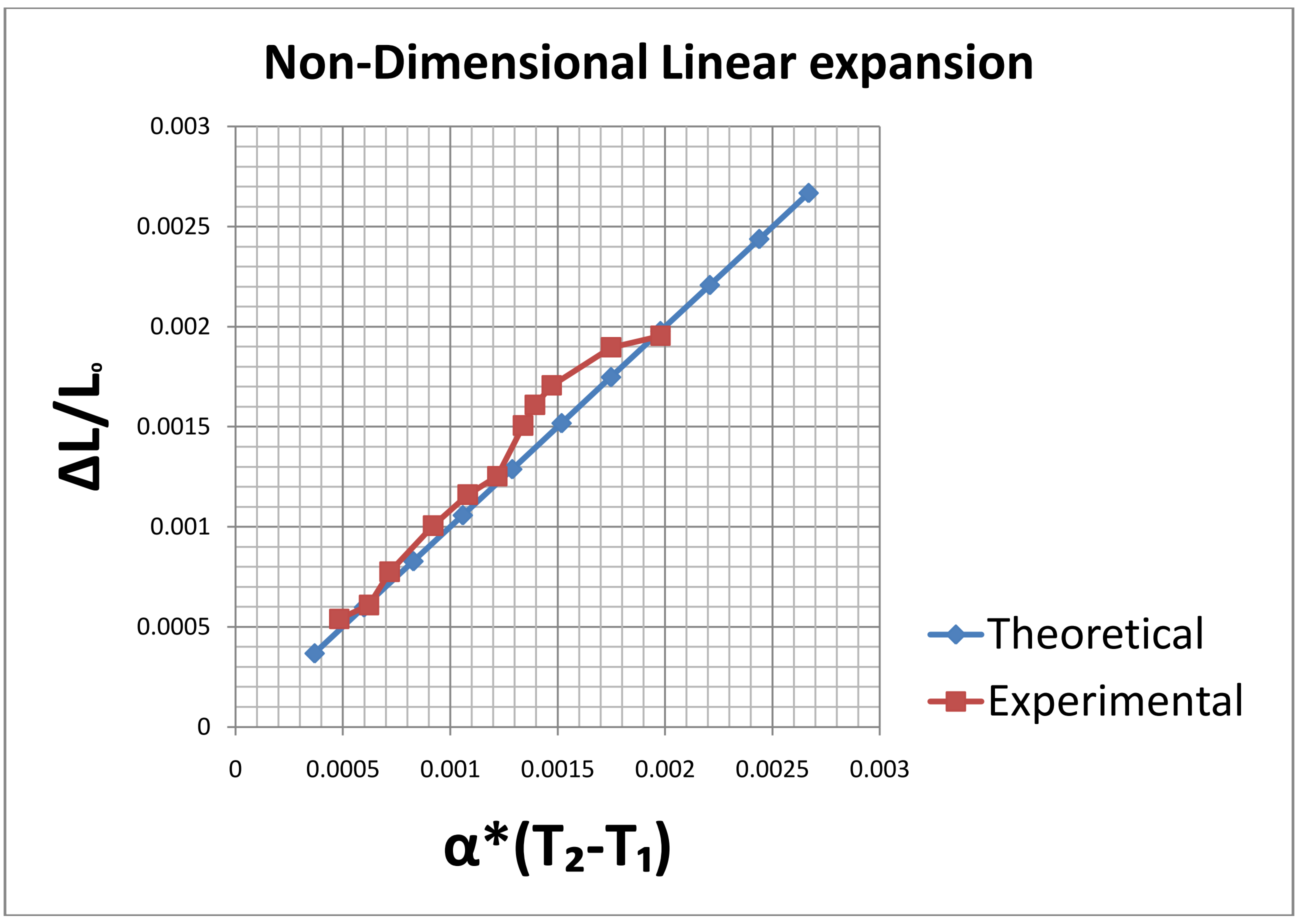


Figure 12: Theoretical & Experimental Data

Experiment 2: Proof of Concept Testing

Before final design and testing a proof of concept test was implemented. The heater and blower that will be implemented were purchased for use with the proof of concept testing. All materials and insulation was selected based on the final design. A thermal chamber was built in order to test the heating process (Figure 13). The unit was powered up and several tests completed. A total of two housings were used. The housing was placed in the chamber and the heater and blower started. The temperature was recorded using a thermo couple. The time was also recorded. The experimental data was then plotted against the MatLab model. Two graphs were generated. The first is a scatter plot showing all the data from all experimental tests (Figure 14). The next graph shows the average data with error bars (Figure 15). Experiment two verifies our Matlab model.

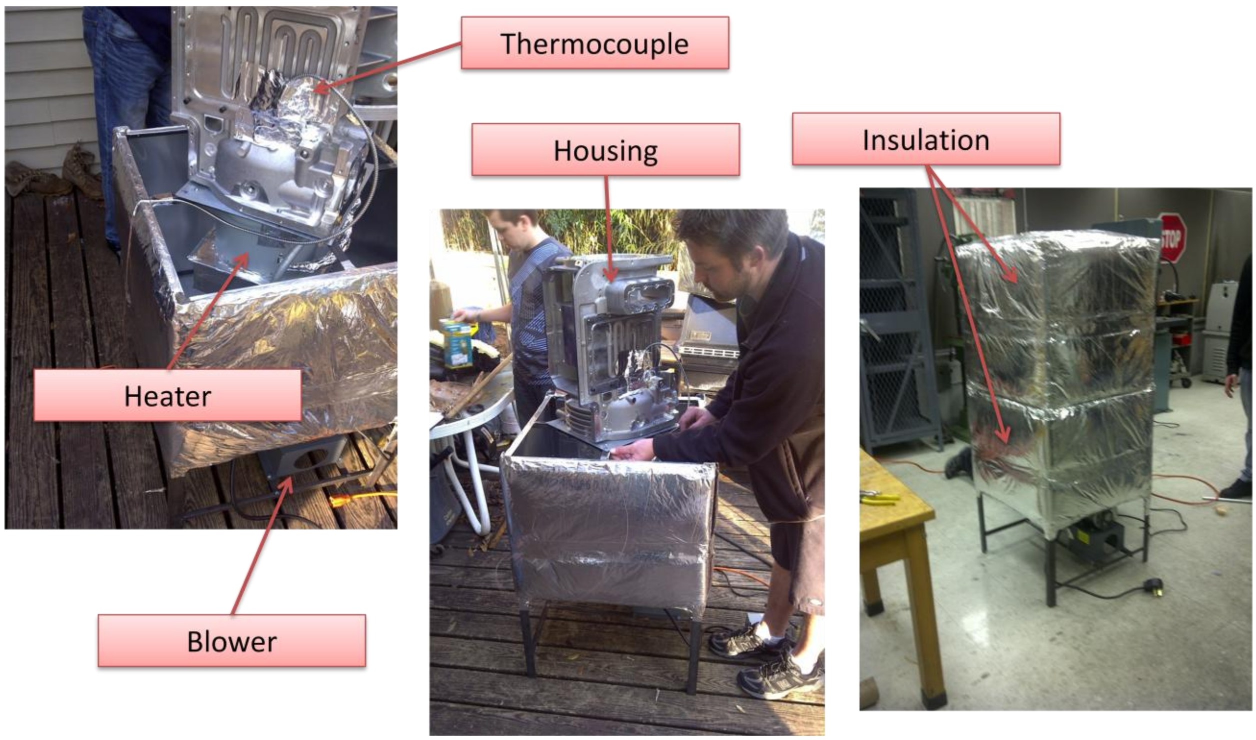


Figure 13: Thermal Chamber

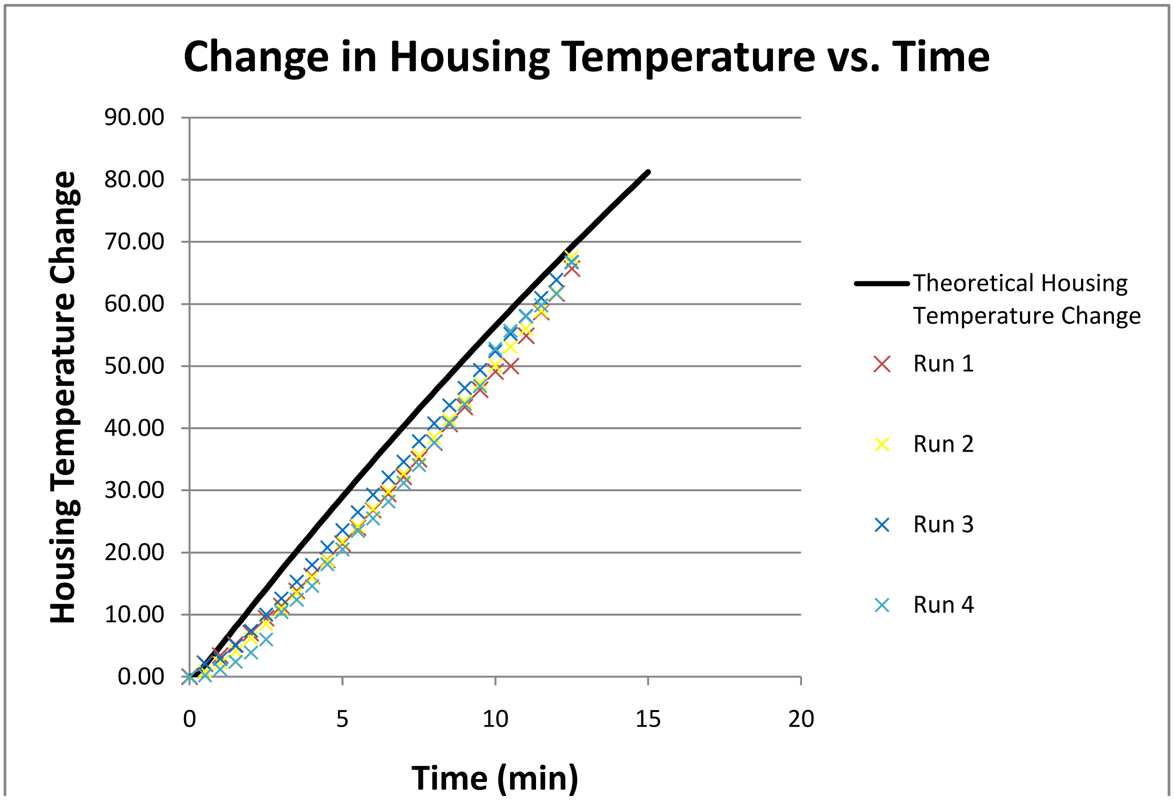


Figure 14: Experiment 2 Data

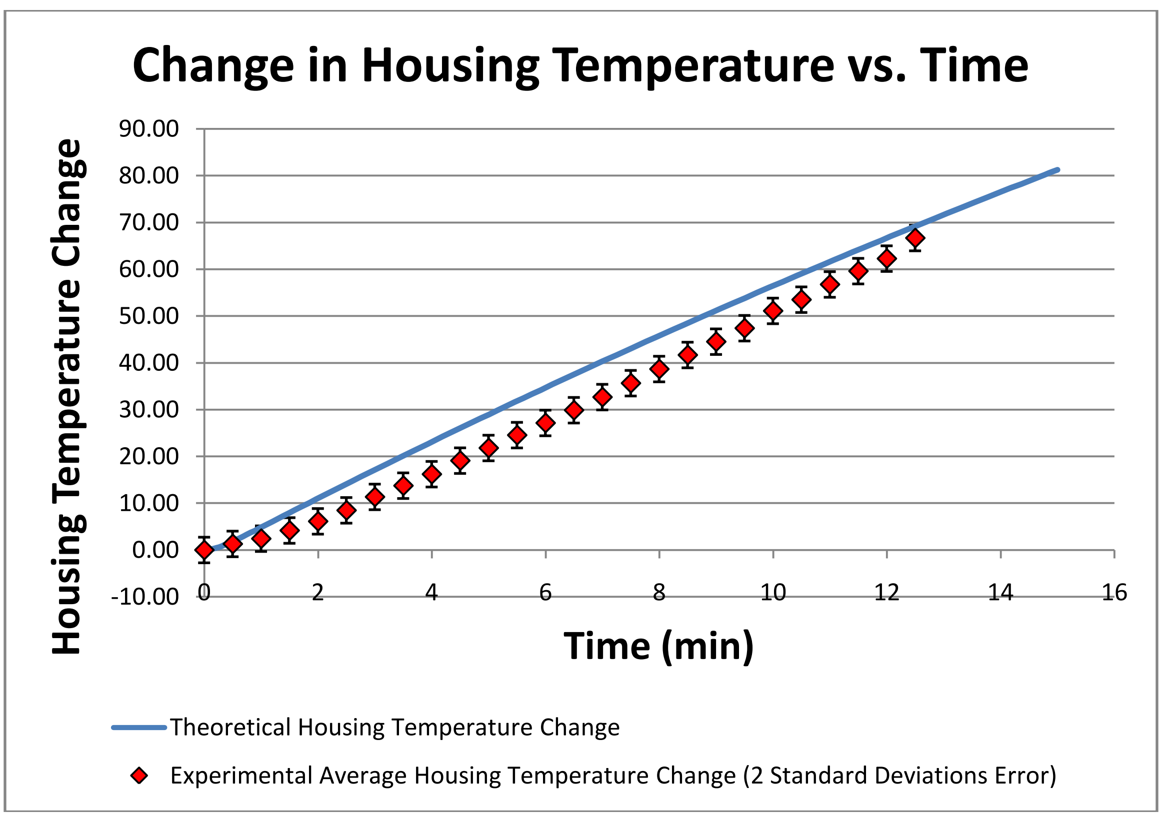


Figure 15: Experiment 2 Average Data

System Optimization

The two critical parameters in our system are the internal convection coefficient and the heater wattage. In order to understand how these factors affect our system we used our Matlab program to plot the system response using various heater powers and convection coefficient (Figure 16).

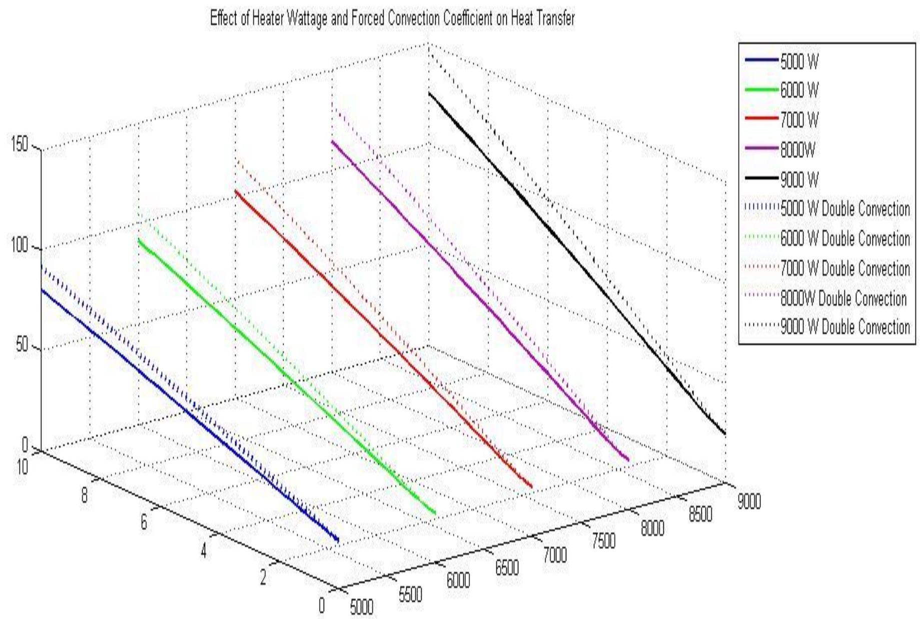


Figure 16: Heater Power & Convection Coefficients

An important trend in the data is that doubling the convection coefficient for a chosen heater size has nearly the same effect as raising the heater wattage by 1000W. Building a cost effective unit requires these properties to be analyzed. It could be possible to simply design a better nozzle or vary the flow rate to increase the convection coefficient. An ideal nozzle would induce turbulence in the flow to enhance heat transfer. The convection coefficient we used for our analysis was based on turbulent flow in a pipe. Our system obeys this initially but as the air passes through the top of the housing it then travels around the outside of the housing it is no longer flow in a pipe. There are no direct correlations for this type of flow. It is quite possible that the actual convection coefficient could be greater than what we calculated. When the flow changes direction it will become more turbulent and perhaps increase the convection coefficient. The only way to verify our model is to create an actual proof of concept and see how it reacts to the heater we have selected. If the initial nozzle doesn't work as expected we have several others to try. The blower we have also has the ability to vary the flow rate.

Increasing the heater size is another possible way to decrease the length of heating time. This is a very effective, but expensive way to increase the heat transfer into the housing. A off the shelf resistive heater bigger than the one we selected

(5600W) will be difficult to come by. We found several companies online that can custom fabricate a specific heater but they are very expensive. This will be a last possible resort for us.

Prototype

The following images show the prototype of the heater station assembly. The overall design consists of the table that is a part of the production line. Below the table top, there are 2 sealed off sections, the central one houses the heater and the shroud. This section has a hot air inlet on the bottom below the heater. After passing through the heater, hot air goes through the shroud and the nozzle and exits out through the table top, heating up the compressor housing. Air is then diverted by the hood back down and through the table slits that allow the air to enter the second sealed chamber, from which it is sucked out by the heater fan and redirected back to the heater in a continuous cycle. Above the table top rests the hood which has openings on two sides to allow the compressor housings to slide in and out during production, as well as an exhaust fan which is utilized during the cooling cycle.

1 Housing
2 Heater
3 Hood
4 Exhaust Fan
5 Table
6 Heater Fan
7 Shroud
8 Nozzle

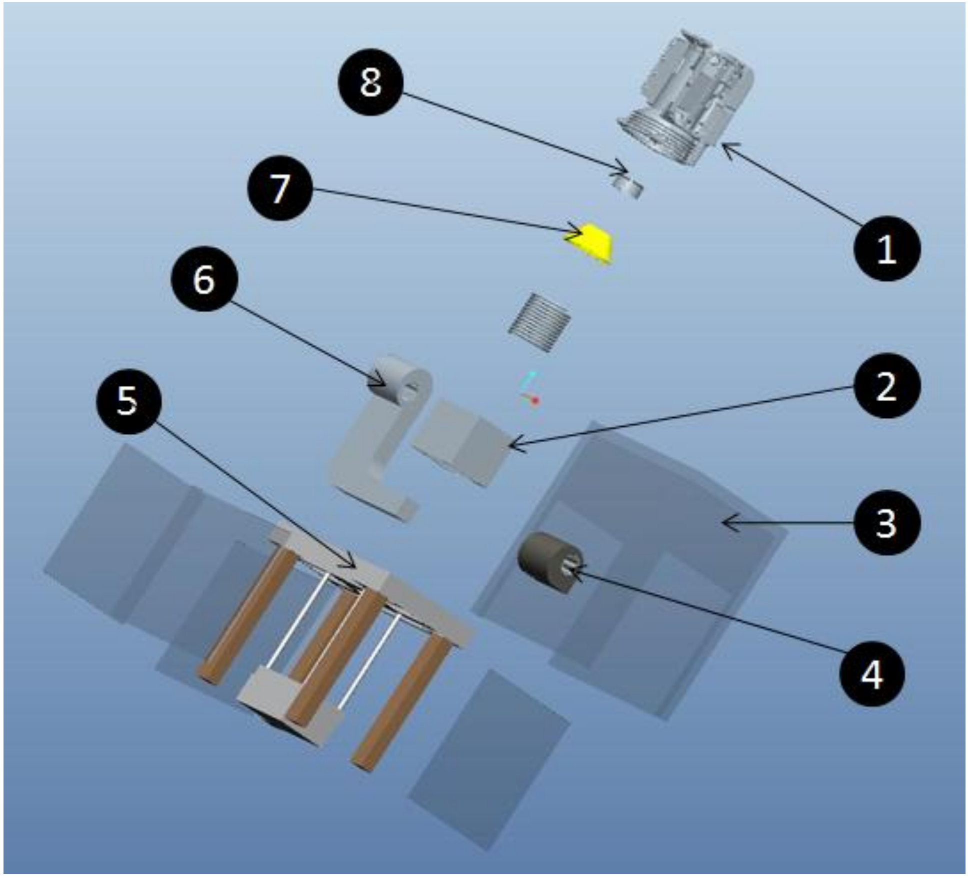


Figure 17: Heater Design Prototype, Exploded View

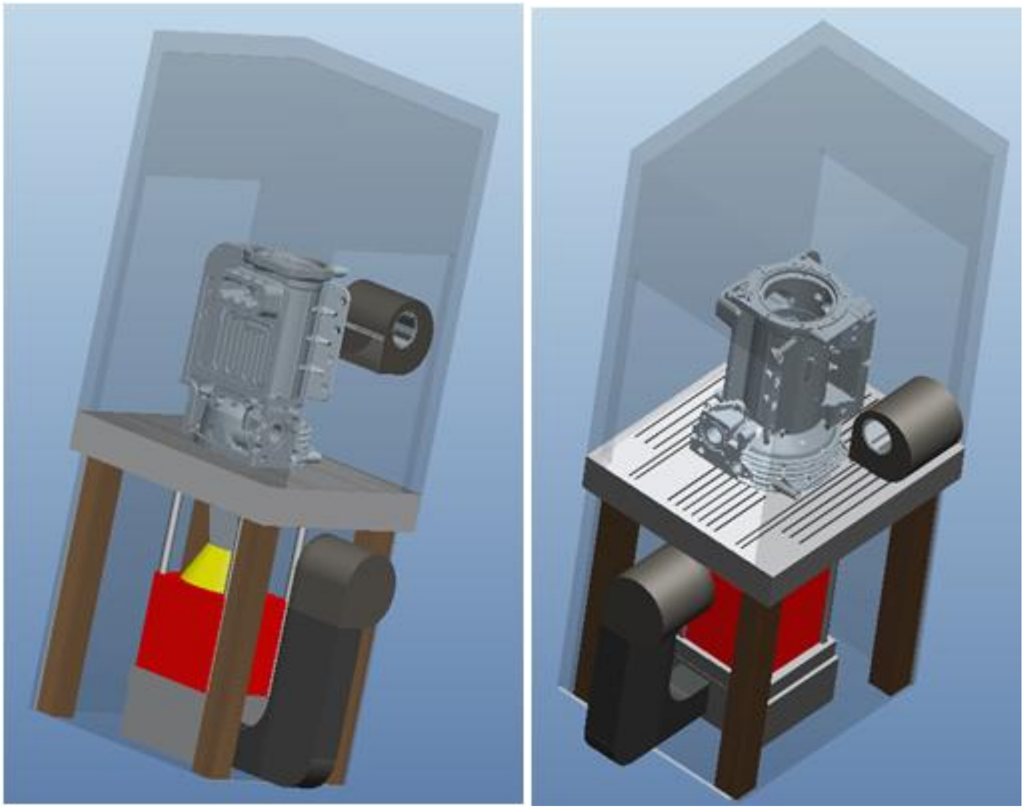


Figure 18: Heater prototype, Assembled view

The heater assembly used in the proof of concept will also be used in the prototype. The final structure will be manufactured from 80/20 engineering extrusion. The final clearance is 60 microns. This will require that the housing and the stator are precisely aligned. To accomplish this, both the base that the housing will sit on and the stator insertion device must be mated together (Figure 19).

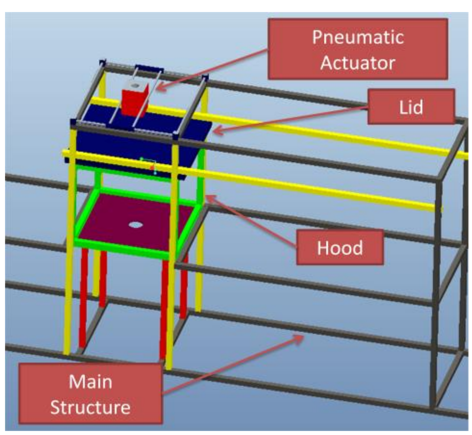
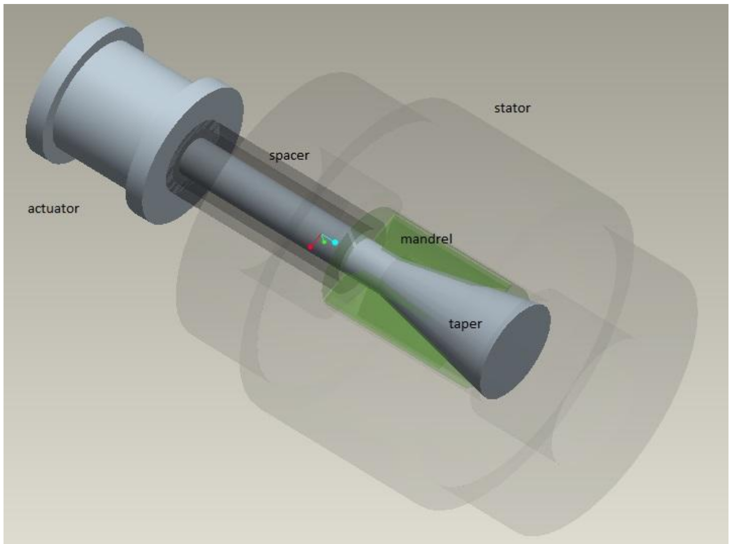


Figure 19: Main Structure

Assembling the entire structure as one piece allows the alignment to be more precise. The actual unit will utilize a pneumatic actuator and expanding mandrel to clamp and hold the stator. Implementation of this device is beyond the scope of the project but is included (Figure 20). All required engineering drawings are included in Appendix III. Details of process function can be found in the Operation Manual in Appendix I.



Environmental, Health & Safety Issues

Environmental

There are no direct negative effects from this design. We are not working with any harmful fluids that require special permits for storage and disposal. Also no special gases are used in the pneumatic system, thus eliminating environmental damage in case of malfunction. No radio signals are generated at all, thus not requiring the use of magnetic shielding. Also none of the systems are vulnerable to electromagnetic interference either.

Health

- Vent
- Crush hazard
- Touch/burn hazard

The most important aspect of the design that has to be accounted for is the hot air escaping in between the individual cycles. It would be preferable to have a clear ceiling or one that is sufficiently high from the design. Also insuring that no pipes or conduits are running overhead is preferable.

The next important aspect is the crush hazard that is always present when working with heavy parts or automated machinery. Because the compressor housing is quite heavy on its own, handling it will have to be extremely careful, and thus this design is focused on minimizing the lifting required. Simply sliding along the surface works best when working with heavy items and that's why the design is being incorporated into the current production line that uses rollers to help with moving the heavy compressor housings. Also the positioning and locking mechanism for the housings is designed to be flush with the working table surface.

Since the use of pneumatic actuators was decided on, another crush hazard is present because of the great amount of force these can generate. But because the mechanism for inserting the stators is high above the working height, human interference is kept out of reach thus making it safer.

Another issue that has to be kept track of is the working temperatures and their effect later on in the production cycle. But because the temperature of the desired expansion is in the 85 degree C area, the housings could easily be handled with regular household gloves as opposed to the industrial ones being used currently. This also benefits the time required to cool the housing back down, minimizing the down time before the production can continue. Since the design heat chamber will be subject to high air temperature as well, adequate insulation is required throughout the support structure.

Safety

- Temperature sensors
- Infrared curtains

To insure the maximum safety of the design and maximum consistency of the results, some precautionary measure will have to be implemented. Since the ambient temperature inside the factory will vary with both seasons and time of day, temperature sensors will have to be implemented to make sure the necessary expansion of the housing is achieved. By simply placing thermocouples on the housings every time will give the best results but will complicate the final design. Otherwise measuring the air temperature in the chamber during heat cycle and going off a general average temperature will be the quickest and simplest way to insure the clearance was achieved to safely insert the stator. As a precautionary measure a thermostat should be incorporated into the heater to ensure the maximum performance and safe operation of the heating elements.

In conjunction to ensure absolutely safe operation of the stator inserter, infrared curtains can be used in front of the operator side of the design, to guarantee that no body parts are present in the working volume of the actuators.

Design for Manufacturability

The machine does not have many complex moving parts so the final concept will not hard to manufacture. The material used for most of the machine is 80/20, which was easy to receive through our sponsor and also, easy to cut, assemble and has optimal material properties. The downside to using 80/20 is it is relatively expensive. Basic tools are needed to assemble 80/20. The material is designed with pre-drilled hole designed for ¼-20 thread tapping. Other materials used for the design are aluminum and steel. The most complex part is the main base of the structure. With the supplied drawings and files the base can be manufactured using a water jet. The final cost is over our design budget, the only way to keep everything aligned properly was by using 80/20 extrusion. This working prototype is functional as far as the design intent. Turbo Corr will be supplied with all the drawings for manufacture. See cost analysis for economic detail.

Table #:Convection Heater Cost Analysis

Part	Description	Unit Price	Quantity	Total Price
Electric Heater	5600 W	$138.09	1	$138.09
Blower	250 CFM	$249.95	1	$249.95
Steel Plate	24"X24"X0.5"	$320.00	1	$320.00
Table Assembly	Hardware	N/A	12	$37.16
Ultra Flex Hose	5' Length 4" ID	$74.24	1	$74.24
80/20 Extrusion	25 Series Mono Slot Bar	$63.40	N/A	$760.80
80/20 Hardware	Misc. Hardware	N/A	N/A	$1,878.85
Sheet Metal	Heater Covering	$43.57	1	$43.57
			Total	$3,502.66

References

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Appendix I: Operation Manual

Operation Manual

General Operation

The operation of this unit is fairly simple, the goal is to heat up the aluminum housing and insert a stator into it. This machine relies on two electrical components that allow this to occur, the blower and the heater (Figure 19).

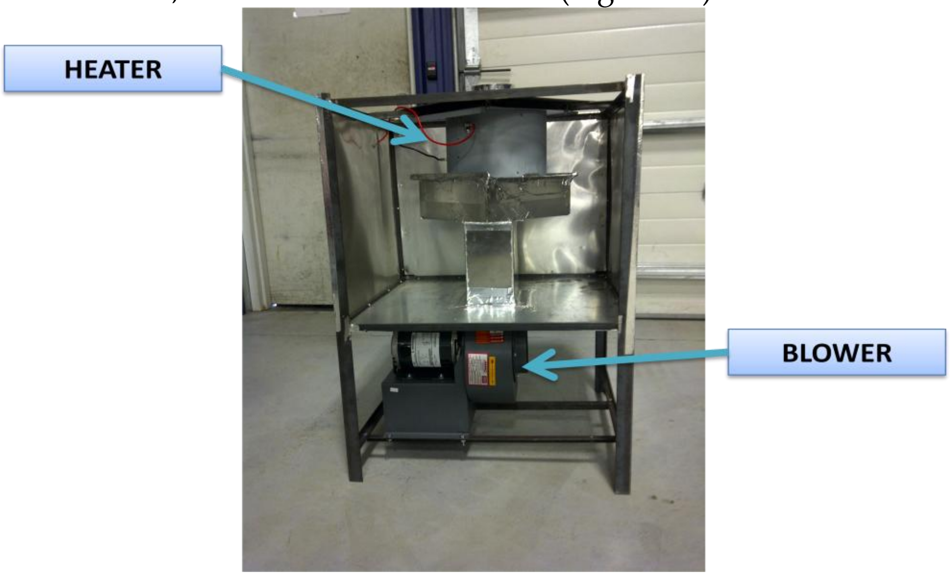


Figure 20: Blower and Heater

These will be varied on/off for different stages in the operation cycle. The machine also relies on a few mechanical components, the hood slides vertically up and down, the lid that slides horizontally left and right (Figure 20).

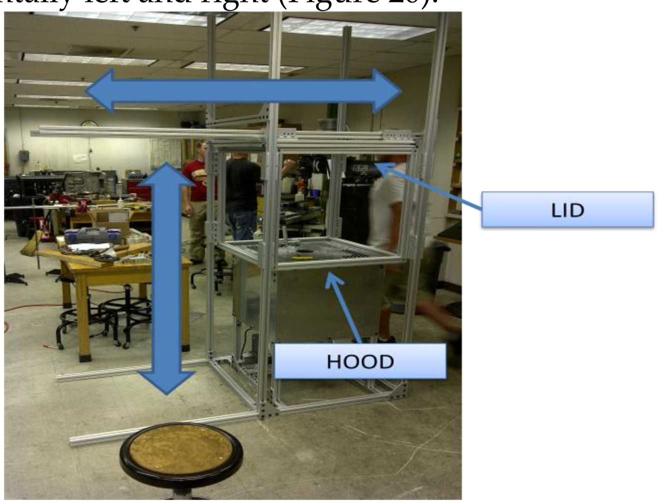


Figure 21: Hood and Lid

The stator insertion device will utilize a pneumatic cylinder coupled with an expanding mandrel to lift and place the aligned stator.

The following steps are utilized to accomplish the goal of stator insertion; first the machine needs to be prepared for operation by turning the heater on with the unit sealed, to seal the unit the hood must be up and the lid to the left in the closed position. Next the system will preheat for minimal losses during an operation cycle. Once pre-heated the unit can be opened up and prepared for the first operation cycle, to open up the unit the hood must be slid down and the lid slid to the right. Next a stator must be prepared for insertion at the end of the cycle.

Stator and Housing Alignment

The main base has a total of eight alignment pins. Four of which are used to align the stator, the remaining four are used to locate the housing (Figure 21).

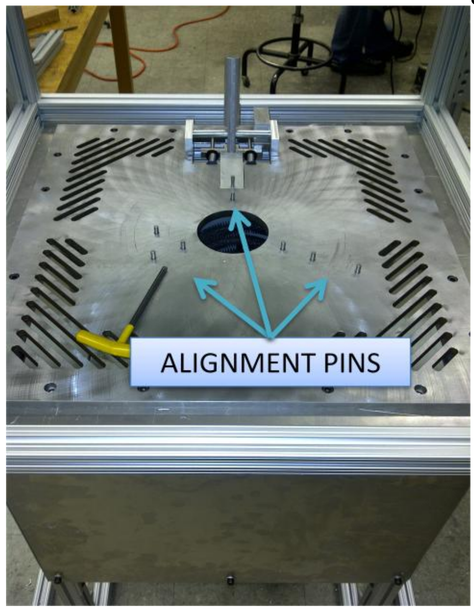


Figure 22: Alignment Pins

Step 1: Alignment Ring

The first step is locating the alignment ring onto the base plate (Figure 22). The ring is then rotated to the fourth pin to locate the ring concentrically and angularly to the base (Figure 23).

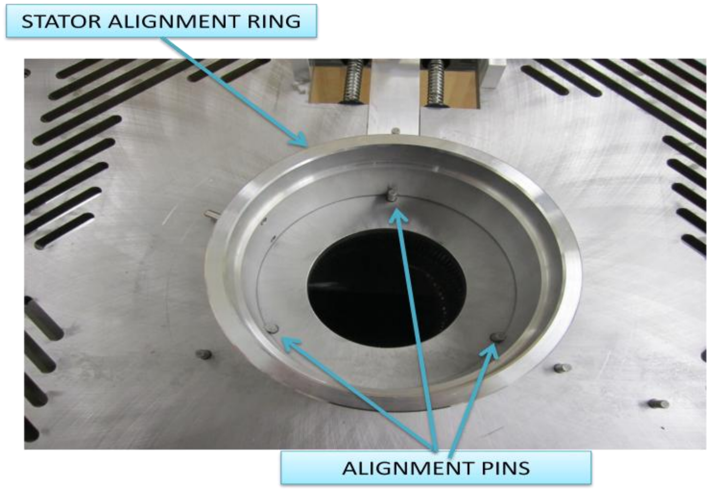


Figure 23: Alignment Ring Concentric Location

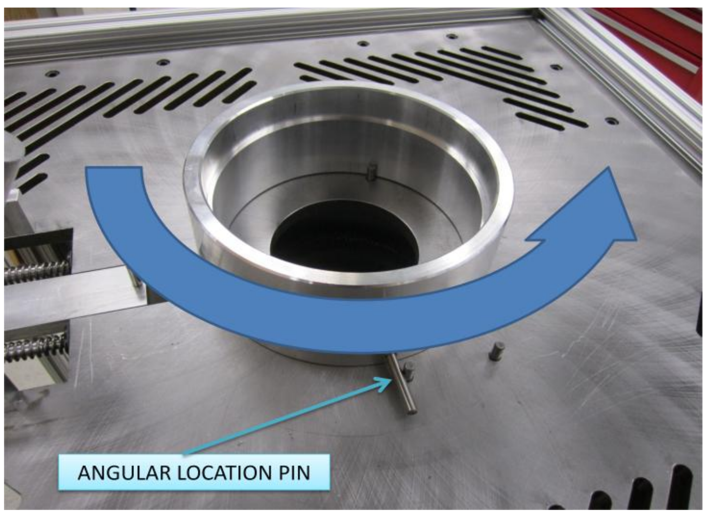


Figure 24: Alignment Ring Angular Location

Step 2: Stator Alignment

The second phase involves placing the stator into the ring (Figure 24). The stator is then rotated until the alignment pin can be inserted through the ring and the stator (Figure 25). This process locates the stator both concentrically and angularly to the ring and hence the base.

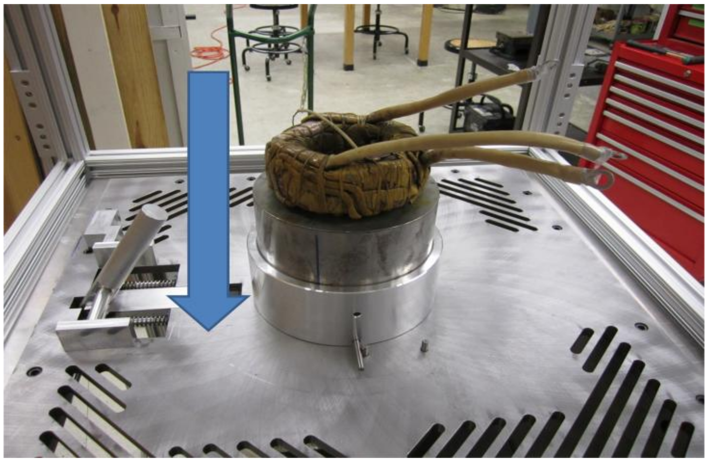


Figure 25: Placing Stator

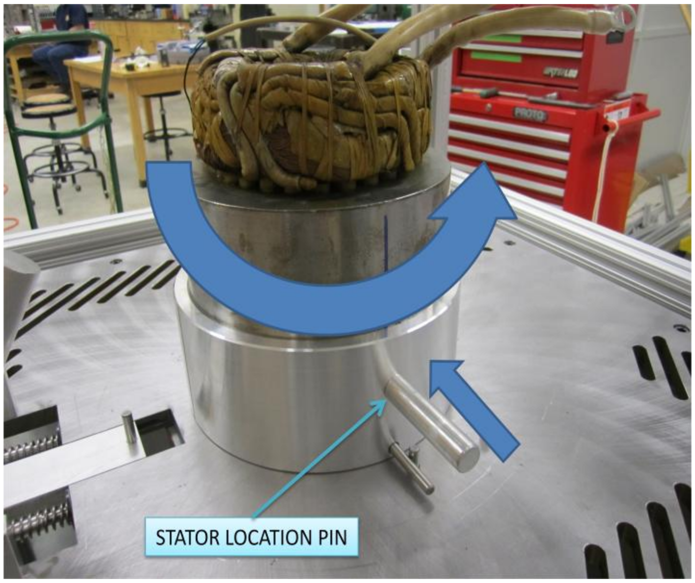


Figure 26: Stator Location Pin

Step 3: Housing Location

The last step is to align the housing concentrically and angularly to the base. This involves moving the handle to the load position and placing the stop to engage the lock (Figure 26). The housing is then placed on the base (Figure 27). The handle is disengaged allowing the three pins to locate the housing concentrically (Figure 28). The housing is then rotated to the fourth pin aligning the housing angularly (Figure 29).

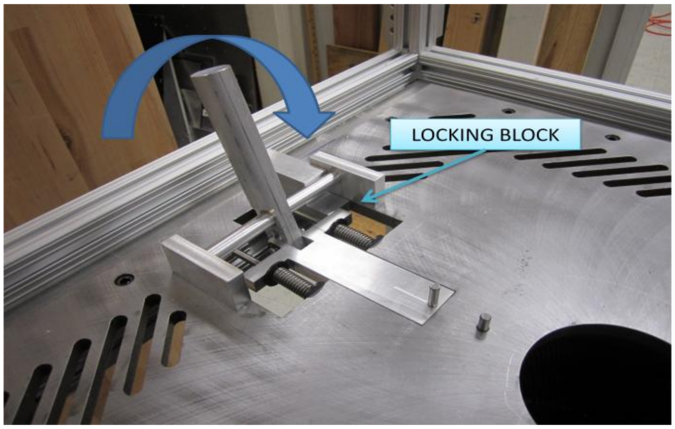


Figure 27: Housing Location Handle and Lock Block

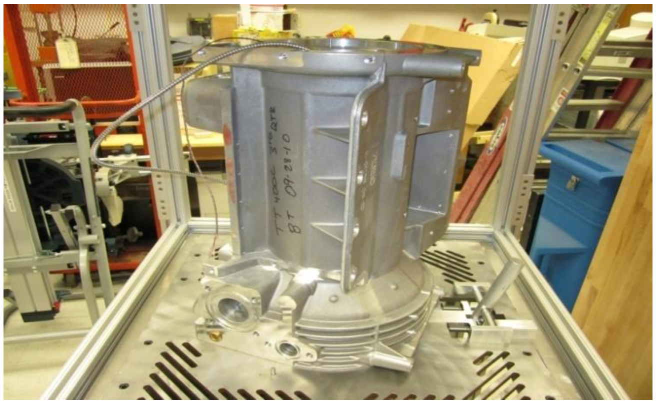


Figure 28: Housing Placement

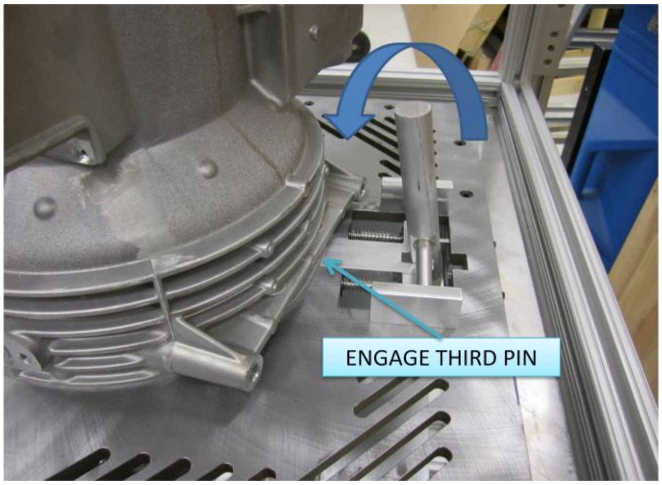


Figure 29: Engaging Housing Lock

Figure 30: Locating Housing Angularly

This ensures that the stator and the housing are located together this completed the alignment process. Once locked in place the hood can be lifted back up and the lid slid over the hood to seal the unit, now the heater and blower can be turned on to begin the heating cycle. Thermocouples and/or other sensors determine when the part has reached its final temperature, once this occurs the blower is shut off, the hood is slid down, and the lid is slid to the right. This opens up the unit giving the stator inserter to access the housing, and can now insert the stator that was picked up in the beginning of the cycle, as mentioned early the operator just has to push a switch to lower it and then push a second switch to drop it and then the first switch again to raise it. Finally the housing can be slid out of the machine and onto the cooling line to complete the interference fit. The process can then be started all over again for another housing and stator.

Appendix II: Calculations

Thermal Expansion

$\Delta L = L_o * \alpha * \Delta T$

$\frac{\Delta L}{L_o} = \alpha * \Delta T$

Variable Definition

$\Delta L = Change\ in\ length$

$L_o = Initial\ length$

$\alpha = Thermal\ expansion\ coefficient$

$\Delta T = Change\ in\ temperature$

Thermal Expansion Sample Code

This code is specific to our particular housing.

```
%Calculating the clearance between the housing and the stator
%based on temperature. Linear thermal expansion of the radius is assumed
%Calculations assume thermal equilibrium within the material
%The calculation of heat needed to reach required
%temperature is based on second law
%Heat required is mC*(t2-t1)
%Specific Heat and Density designated 1 are for pure aluminum
%Specific Heat and Density designated 2 are for 2024 aluminum


clear all;
%Smallest possible part diameter based on drawing
Dinit = .171450;
%Maximum stator size
Stat = .171630;
%Linear expansion coeffient for aluminum (units m/m*degC)
alpha = 0.000023;
r2 = 0.230;
r1 = 0.171450;
h = 0.20065;
V = 0.012688855;
Tinit = 25;
Rho1 = 2670;
```

```
C1 = 963;
Q1 = zeros(1,201);
TfinalF = zeros(1,201);
TfinalC = zeros(1,201);
LDelta = zeros(1,201);
Linit = pi*Dinit;
DDelta = 0;
TDelta = 0;
Dnew = zeros(1,201);
Cler = zeros(1,201);
i = 0;
for t = 25:0.5:125
   i = i + 1;
   TDelta = t-Tinit;
   LDelta = alpha*Linit*TDelta;
   DDelta = LDelta/pi;
   Dnew = (Dinit + DDelta);
   Cler(i) = (Dnew-Stat)*10^6;
   TfinalF(i) = (9/5)*t+32;
   TfinalC(i) = t;
   Q1(i) = (Rho1*V*C1*(t-Tinit))/1000;

end
figure(1)

plot(TfinalF,Cler),title('Clearance Versus Temperature'),
xlabel('Temperature (Deg F)'),ylabel('Clearance (Microns)'),grid on

figure(2)

plot(TfinalC,Cler),title('Clearance Versus Temperature'),
xlabel('Temperature (Deg C)'),ylabel('Clearance (Microns)'),grid on

figure(3)

plot(TfinalC,Q1),title('Heat Versus Temperature (Aluminum A356 T6)'),
xlabel('Temperature (Deg C)'),ylabel('Heat (kJ)'),grid on

figure(4)

plot(TfinalF,Q1),title('Heat Versus Temperature (Aluminum A356 T6)'),
xlabel('Temperature (Deg F)'),ylabel('Heat (kJ)'),grid on
```

Heat Input Sample Code

This code allows the user to input a desired clearance in mcrons. The program returns the final temperature and the needed heat input.

```
%Calculating the temperature needed to reach desired clearance
%between the housing and the stator.
%Linear thermal expansion of the radius is assumed
%Calculations assume thermal equalibrium within the material
%Calculating the Heat needed to reach desired Temperature
%Treating the housing as a cylinder


clear all;
%Smallest possible part diameter based on drawing
Dinit = .171450;
%Maximum stator size
Stat = .171630;
%Linear expansion coeffient for aluminum (units m/m*degC)
alpha = 0.000023;
CL = input('Input desired clearance in microns >');
TI = input('Input initial temperature in degrees C >');
Rho = 2670;
C = 963;


DNew = (CL*10^-6)+Stat;
DeltaD = DNew - Dinit;
DeltaT = DeltaD/(Dinit*alpha);
TC = DeltaT + TI;
TF = (9/5)*TC+32;

r2 = 0.21910;
r1 = 0.171450;
V = 0.012688855;
Heat = (Rho*V*C*(TC-TI))/1000;



sprintf('Required Temperature is %.2f deg F',TF)
sprintf('Required Temperature is %.2f deg C',TC)
sprintf('Required Energy is %.2f kJ',Heat)
```

Internal Convection Coefficient

$\rho = \frac{P}{RT}$

$Re = \frac{\rho V D_h}{\mu}$

$Nu = 0.023 Re^{0.8} Pr^{n}$

$h_i = \frac{Nuk}{D_h}$

Variable Definitions

$P = Static\ Pressure$

$R = Gas\ Constant$

$T = Tempertaure$

$V = Velocity$

$D_h = Hydraulic\ Diameter$

$\mu = Dynamic\ Viscocity$

$Pr = Prandlt\ Number$

$n = Cooling\ Exponent$

$k = Thermal\ Conductivity$

Natural Convection Coefficient

$Gr = \frac{g\beta (T_s - T_\infty) L_c^3}{v^2}$

$Ra = GrPr$

$Nu = 0.59 Ra^{\frac{1}{4}}$

$$h_o = \frac{Nuk}{L_c}$$

Variable Definitions

$g = Gravitational\ Constant$

$\beta = Volume\ Expansion\ Coefficient$

$T_s = Surface\ Temperature$

$T_\infty = Ambient\ Temperature$

$L_c = Characteristic\ Length$

$v = Kinematic\ Viscocity$

$Pr = Prandlt\ Number$

Thermal Resistance Network

$$R_i = \frac{1}{h_i A_{heated}}$$

$$R_w = \frac{t}{k_{insula\ tion}\ A_{heated}} + \frac{t}{k_{metal}\ A_{heated}}$$

$$R_o = \frac{1}{h_o A_{heated}}$$

$$R_{tot} = R_i + R_w + R_o$$

$$\dot{Q}_{loss} = \frac{(T_2 - T_\infty)}{R_{tot}}$$

$T_2 = Hood\ Surface\ Temperature$

$T_\infty = Temperature\ of\ Surrounding\ Air$

$A_{heated} = Heated\ Housing\ Surface\ Area$

$t = Hood\ Material\ Thisckness$

$k_{hood} = Hood\ Material\ Thermal\ Conductivity$

$h_o = Outside\ Natural\ Convection\ Coefficient$

$h_i = Inside\ Convection\ Coefficient$

Control Volume Analysis

System 1 (Housing)

$$m_1 C_1 \frac{dT_1}{dt} = \dot{Q}_{21}$$

$$\dot{Q}_{12} = h_i A_1 (T_2 - T_1)$$

$$\frac{dT_1}{dt} = \frac{h_i A_1 (T_2 - T_1)}{m_1 C_1}$$

System 2(Air in system)

$$m_2 C_v \frac{dT_2}{dt} = \dot{W} - \dot{Q}_{12} - \dot{Q}_{loss}$$

$$\dot{W} = Heater\ Power$$

$$\frac{dT_2}{dt} = \frac{\dot{W}}{m_1 C_v} - \frac{h_i A_1 (T_2 - T_1)}{m_2 C_v} - \frac{(T_2 - T_\infty)}{R_{tot}}$$

Final Coupled ODE System

$$\frac{dT_1}{dt} = AT_1 - AT_2$$

$$\frac{dT_2}{dt} = BT_1 - CT_2 + D$$

$$A = \frac{h_i A_1}{m_1 C_1}$$

$$B = \frac{h_i A_1}{m_2 C_v}$$

$$C = \frac{h_i A_1}{m_2 C_v} + \frac{1}{m_2 C_v R_{tot}}$$

$$D = \frac{\dot{W}}{m_2 C_v} + \frac{T_\infty}{m_2 C_v R_{tot}}$$

Variable Definitions

$h_i = Inside\ Convection\ Coefficient$

$A_1 = Heated\ Housing\ Surface\ Area$

$m_1 = Mass\ of\ Housing$

$C_1 = Specific\ Heat\ of\ Aluminum\ Housing$

$m_2 = Mass\ of\ Air$

$C_v = Specific\ Heat\ of\ Air\ at\ Constant\ Volume$

$R_{tot} = Total\ Thermal\ Resistamce$

$\dot{W} = Heater\ Power$

$T_\infty = Temperature\ of\ Surrounding\ Air$

$A_{heated} = Heated\ Housing\ Surface\ Area$

$t = Hood\ Material\ Thisckness$

$k_{hood} = Hood\ Material\ Thermal\ Conductivity$

$h_o = Outside\ Natural\ Convection\ Coefficient$

Control Volume Analysis Sample Code

Initial System

This first code was used to evaluate all of the parameters for our system. All of the important variables are commented within the program.

```
function Tprime = HeatMaster(t,T)
%Estimating the time to heat the housing varying the heater wattage and
%the insulation of the hood.  This code was also used to determine
%film temperature to evaluate the necessary properties and determine
%convection coefficients


%System 1 (Housing) Properties
m1 = 32;              %mass
C1 = 900;             %Specific Heat
```

```
d2 = 0.2191;          %outer diameter
d1 = 0.17145;         %inner diameter
h = 0.5;              %height
A1 = pi*d2*h+pi*d1*h;  %heat transfer area

%System 2 (Air) Properties
%Control Volume
Lcv = 0.5;              %Length
Wcv = 0.5;              %Width
Hcv = 0.5;              %Height
Vcv = Lcv*Wcv*Hcv;        %Volume
%Hood Volume
L = 0.7;                %Length
W = 0.7;                %Width
H = 1.0;                %Height
Vhood = L*W*H;          %Hood Volume
%Duct Volume
Ld = 1;                 %Duct Length
Wd = 0.5;               %Duct Width
Vduct = Ld*(Wd^2);        %Duct Volume
%Cabinet Volume
Wc = 0.5;               %Cabinet Width
Hc = 1.0;               %Cabinet Height
Vcab = Hc*(Wc^2);         %Cabinet Volume
Vtotal = Vcv+Vhood+Vduct+Vcab;  %Total Volume of Air in system
%Air Properties
Pr = 101000;            %Static Pressure
Tc = 225;               %Air Temp                  VARIABLE
Tk = Tc+273;            %Temp Kelvin
R = 287;                %Gas Constant
m2 = (Pr*Vtotal)/(R*Tk);    %Gas mass
Cp = 1013;              %Specific heat Cp             VARIABLE
Cv = Cp-R;              %Specific heat Cv

%Hood Properties
t = 0.1;                %Thickness
Lh = L;                 %Length of 1 side
Wh = W;                 %width of 1 side
Hh = H;                 %Height of 1 side
Ah = 4*(Hh*Wh)+(Lh*Wh);     %Heat transfer surface area
kh = 0.42;              %Thermal Conductivity hood      VARIABLE

%Inside Convection Properties
Qus = 250;              %Volumetric flow (US)          VARIABLE
Qsi = Qus*0.0004719;       %Volumetric flow (SI)
rho = (Pr/(R*Tk));         %Density
Nd = 0.1;               %Nozzle Diameter              VARIABLE
```

```
Dh = 0.132;             %Hydraulic Diameter         VARIABLE
V = Qsi/((pi*Nd^2)/4);  %Velocity
mu = 2.76*10^-5;        %Dynamic Viscosity @250C     VARIABLE
Pr = 0.6946;            %Prandlt       @250C      VARIABLE
ka = 0.04104;           %Thermal Cond.    @250C      VARIABLE
Re = (rho*V*Dh)/mu;     %Reynolds #
n = 0.3;                %Cooling Exponent
Nuf = 0.023*(Re^0.8)*Pr^n;  %Nusselt Dittus-Boeler corr.
hi = (Nuf*ka)/Dh;       %Convection Inside

%Outside Convection Properties
Tinf = 25;              %Temp far away
Tsurf = 121;            %Surface Temp               VARIABLE
Tfilm = (Tinf+Tsurf)/2; %Film Temp
TfilmK = Tfilm+273;     %Film temp K
g = 9.80;               %Gravity constant
beta = 1/TfilmK;        %Beta
Lc = Hh;                %Characteristic Length
vn = 1.995*10^-5;       %Kinemetic visc.  @Tfilm      VARIABLE
Prn = 0.7177;           %Prandlt       @Tfilm      VARIABLE
kn = 0.02881;           %Thermal Cond.    @Tfilm      VARIABLE
Gr = (g*beta*(Tsurf-Tinf)*Lc^3)/(vn^2); %Graschoff
Ra = Gr*Prn;            %Raleigh Number
Nun = 0.1*Ra^(1/3);     %Nusselt natural
ho = (Nun*kn)/Lc;       %Convection inside



%Resistance Network
Ri = 1/(Ah*hi);         %Inside Resistance
Rw = t/(kh*Ah);         %Wall Resistance
Ro = 1/(Ah*ho);         %Outside Resistance
Rtot = Ri+Rw+Ro;        %Total Resistance

%Heater Properties
W = 5200;               %Heater Power

%Simplifications
A = (hi*A1)/(m1*C1);
B = (hi*A1)/(m2*Cv);
C = ((hi*A1)/(m2*Cv))+(1/(m2*Cv*Rtot));
D = (W/(m2*Cv))+(Tinf/(m2*Cv*Rtot));

Tprime = [A*T(2)-A*T(1);B*T(1)-C*T(2)+D];
```

Solving the initial system

```
%This Program uses the coupled system of Ode
%to generate temperature profiles for the housing and air temperatures
clear all;

T0 = [25,25];
tspan = [0,600];
[t,T] = ode45(@HeatMaster,tspan,T0);
Tmin = t/60;
figure(1)
plot(Tmin,T(:,1),'r')
figure(2)
plot(Tmin,T(:,2),'b')
```

Control Volume Analysis Parameter System

This code allows variables to be passed to the ODE to see how the system reacts to changing variables.

```
function Tprime = HeatA(t,T,p)
%Estimating the time to heat the housing varying the properties
%listed as Variables
%VARIABLES
hi = p(1);      %Inside Convection          (37.2)
ho = p(2);      %Outside Convection         (4.89)
m2 = p(3);      %Air Mass                   (0.7879)
Cp = p(4);      %Specific Heat Air          (1013)
C1 = p(5);      %Specific Heat Aluminum     (900)
Ah = p(6);      %Hood Surface Area          (3.2900)
t = p(7);       %Hood Thickness             (0.1)
k = p(8);       %Hood Thermal Conductivity  (0.42)
W = p(9);       %Heater Wattage             (4000)

%CONSTANTS
A1 = 0.6135;    %Housing Area
R = 287;        %Gas Constant
m1 = 32;        %Housing Mass
Tinf = 25;      %Temperature Far Away

%CALCULATIONS
Cv = Cp-R;           %Specific Heat constant volume
Ri = 1/(hi*Ah);      %Inside Resistance
Ro = 1/(ho*Ah);      %Outside Resistance
Rw = t/(k*Ah);       %Wall Resistance
Rtot = Ri+Ro+Rw;     %Total Resistance
```

```
%Simplifications
A = (hi*A1)/(m1*C1);
B = (hi*A1)/(m2*Cv);
C = ((hi*A1)/(m2*Cv))+(1/(m2*Cv*Rtot));
D = (W/(m2*Cv))+(Tinf/(m2*Cv*Rtot));

Tprime = [A*T(2)-A*T(1);B*T(1)-C*T(2)+D];
```

Solving the variable system.

```
%This Program uses the coupled system of Ode
%to generate temperature profiles for the hoousing and air temperatures
%the p vector passes the variables to the ODE
clear all;

T0 = [25,25];
tspan = [0,980];                    %Time in seconds
p = [37.2 4.89 0.7879 1013 900 3.29 0.1 0.42 5200];
%Parameters
%[A B C D E F G H I]
%A = Inside convection coefficient        37.2
%B = Outside convection coefficient       4.89
%C = Air mass                             0.7879
%D = Specific heat of air                 1013
%E = Specific heat of aluminum            900
%F = Hood surface area                    3.29
%G = Hood thickness                       0.1
%H = Hood thermal conductivity            0.42
%I = Heater Wattage                       4000
[t,T] = ode45(@HeatA,tspan,T0,[],p);
Tmin = t/60;
%PLOTS

figure(1)
plot(Tmin,T(:,1),'r'),
title('Housing Temperature'),xlabel('Time (min)'),
ylabel('Temperature (C)'),grid on

figure(2)
plot(Tmin,T(:,2),'b'),title('Air Temperature'),xlabel('Time (min)'),
ylabel('Temperature (C)'),grid on

figure(3)
plot(Tmin,T(:,1),'r',Tmin,T(:,2),'b'),title('System Temperature'),
xlabel('Time (min)'),ylabel('Temperature (C)'),grid on,
```

```
legend('Housing','Air','location','SouthOutside')
```

Appendix II: Property Tables

Table 2: Housing Properties

System 1 Properties (Housing)		
Property	**Unit**	**Value**
Mass	kg	32
Specific Heat	m	900
Outer Diameter	m	0.2191
Inner Diameter	m	0.17145
Height	m	0.5
Surface Area	m^2	0.6135

Table 3: Air Volume Properties

System 2 Air Volume Properties		
Property	**Unit**	**Value**
Heater		
Length	m	0.5
Width	m	0.5
Height	m	0.5
Heater Volume	m^3	0.125
Hood		
Length	m	0.7
Width	m	0.7
Height	m	0.7
Hood Volume	m^3	0.49
Duct		
Length	m	1
Width	m	0.5
Duct Volume	m^3	0.25
Cabinet		
Height	m	0.5
Width	m	1
Total Volume	m^3	0.25

Table 4: Air Properties at 250 deg C

System 2 Properties (Air) Evaluated at 250°C		
Property	Unit	Value
Static Pressure	kPa	101
Air Temp	°C	225
Air Temp	K	498
Gas Constant	J/kgK	287
Mass	kg	0.7879
Specific Heat (Constant Pressure)	J/kgK	1033
Specific Heat (Constant Volume)	J/kgK	746

Table 5: Convection Properties

Inside Convection Properties Evaluated at 250°C		
Flow Rate	CFM	250
Flow Rate	m^3/s	0.118
Density	kg/m^3	0.7067
Nozzle Diameter	m	0.1
Hydraulic Diameter	m	0.132
Velocity	m/s	15.02
Dynamic Viscosity	kg/ms	2.76E-05
Prandtl	N/A	0.6946
Thermal Conductivity	W/mK	0.041
Reynolds	N/A	1.76E+04
Cooling Exponent	N/A	0.3
Nusselt	N/A	51.28
Forced Convection Coefficient	W/m^2K	15.94

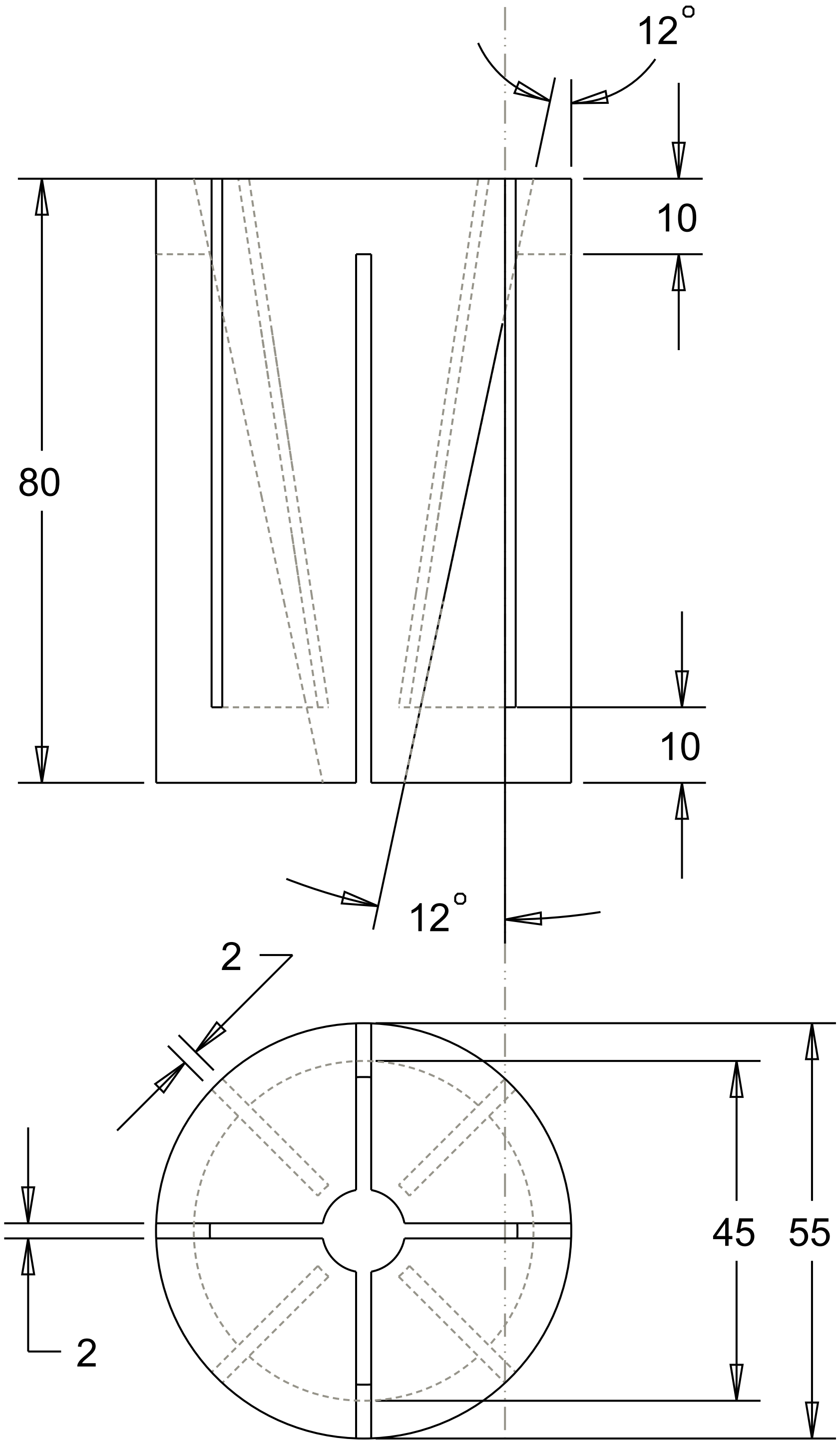
Table 6: Natural Convection Properties

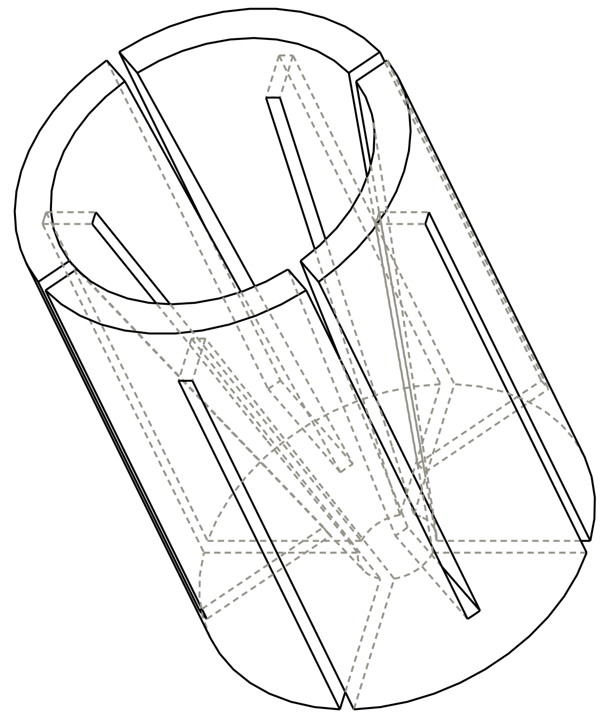
Natural Convection Properties Evaluated at Film Temperature (73°C)		
Temperature Far Away	°C	25
Temperature Surface	°C	121
Film Temperature	°C	73
Film Temperature	K	346
Gravitational Constant	m/s^2	9.8
Beta	1/K	0.0029
Characteristic Length	m	1
Kinematic Voscocity	m^2/s	2.00E-05
Prandtl	N/A	0.7177
Thermal Conductivity	W/mK	0.02881
Graschoff	N/A	6.83E+09
Raleigh	N/A	4.90E+09
Nusselt	N/A	169.88
Natural Convection Coefficient	W/m^2K	4.49

Table 7: Resistance Network Values

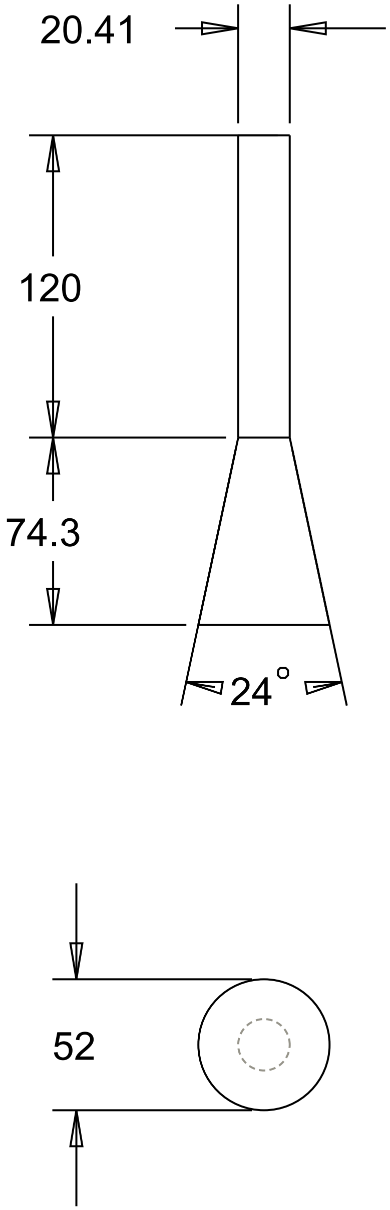
Resistance Network		
Hood Properties		
Thickness	m	0.1
Length	m	0.7
Width	m	0.7
Height	m	1
Hood Surface Area	m^2	3.29
Thermal Conductivity	W/mK	0.42
Inside Resistance	K/W	0.019
Wall Resistance	K/W	0.072
Outside Resistance	K/W	0.062
Total Resistance	K/W	0.154

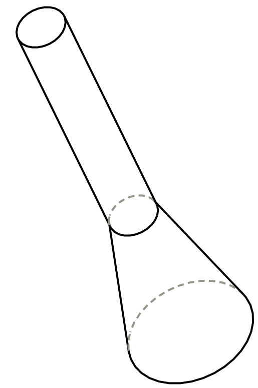
Appendix III: Drawings



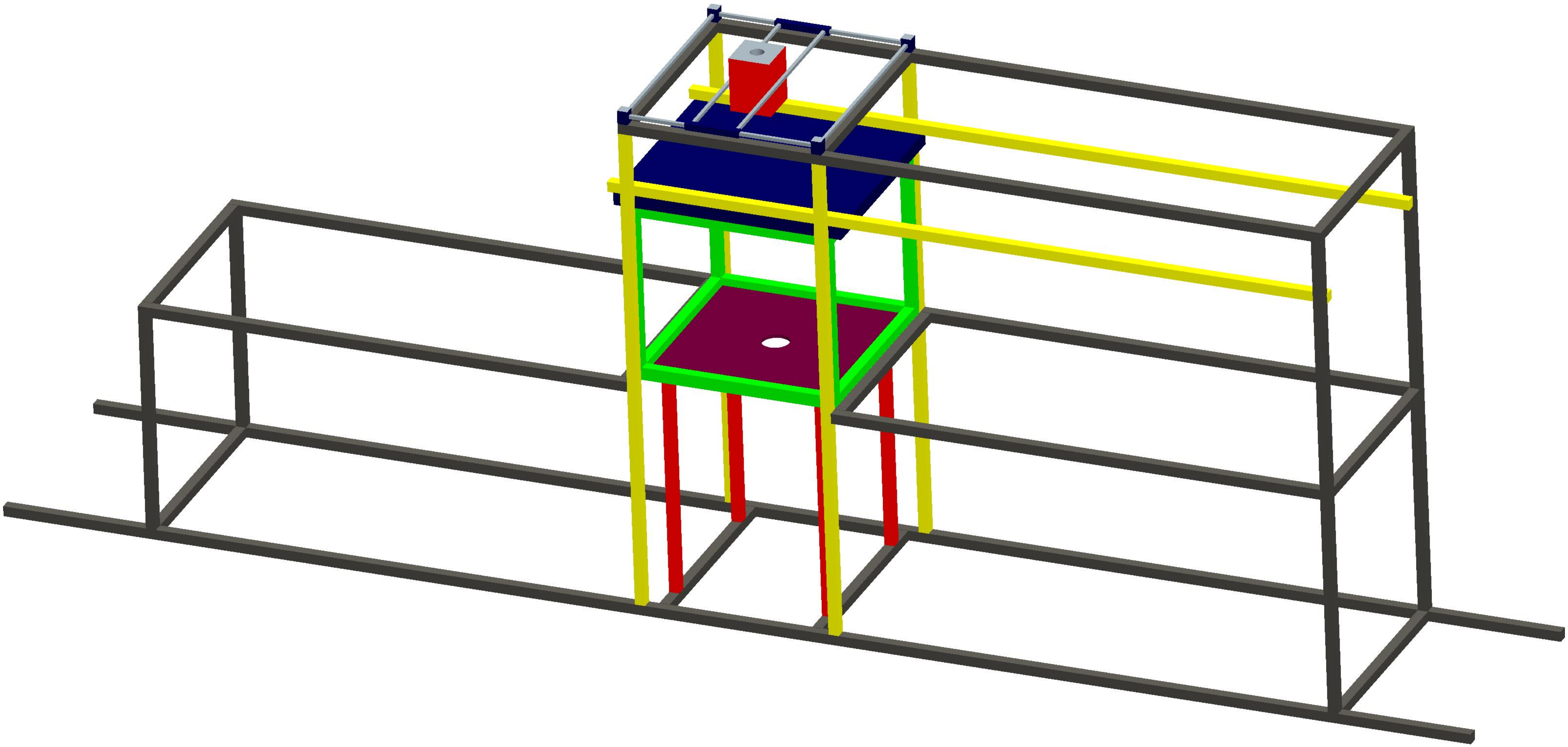


Name	Ivan Dudyak
Part	12 degree Mandrel
Material	Steel
Units	mm





Name	Ivan Dudyak
Part	12 deg taper
Material	aluminum
Units	mm



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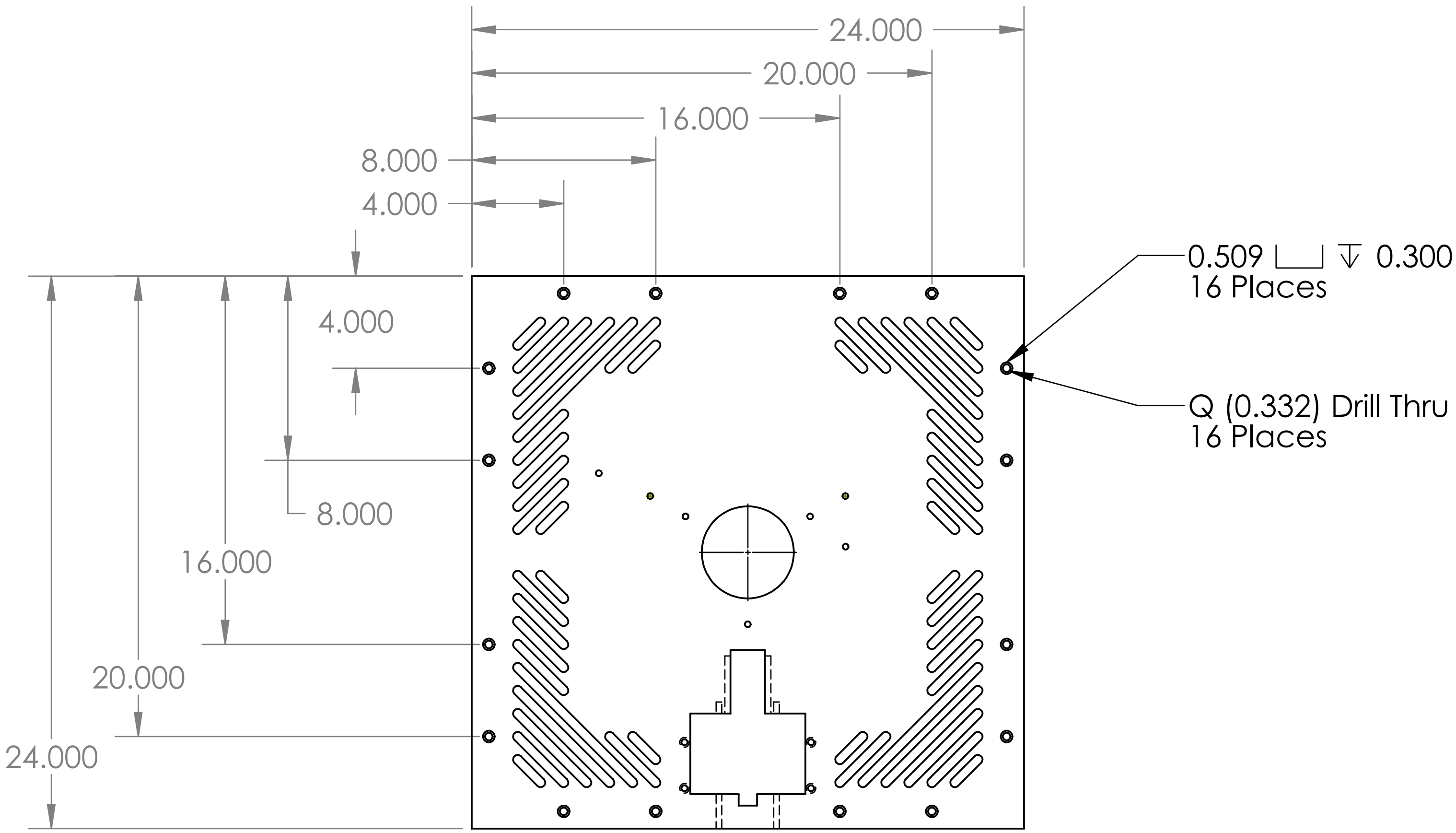
Bill of Materials 80/20	
Length (in)	**Quantity**
27	8
25	4
30.5	6
34	4
24	2
21	6
30.1875	6
70	4
96	4



This Sheet is for overall size
and mounting hole locations

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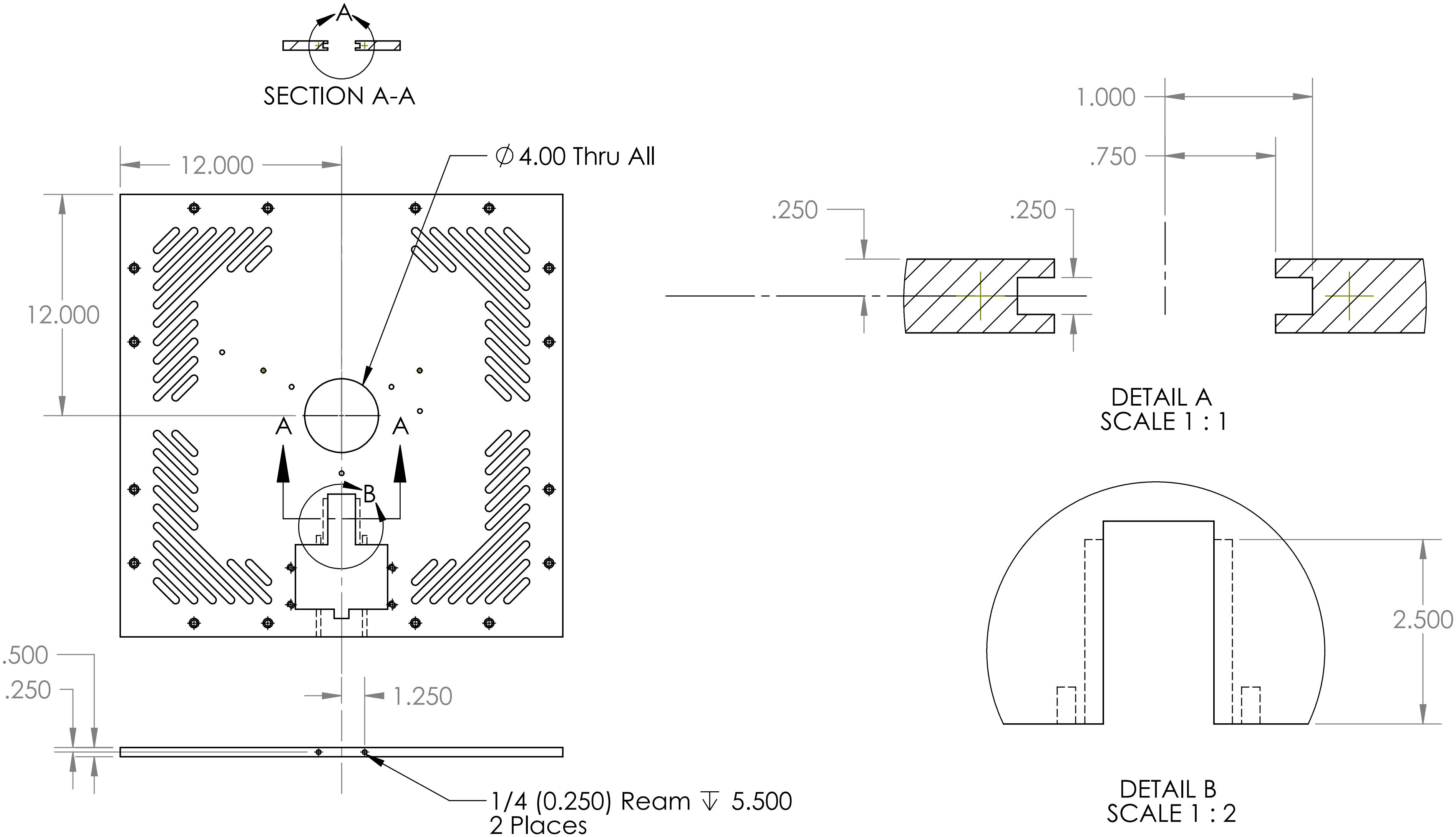

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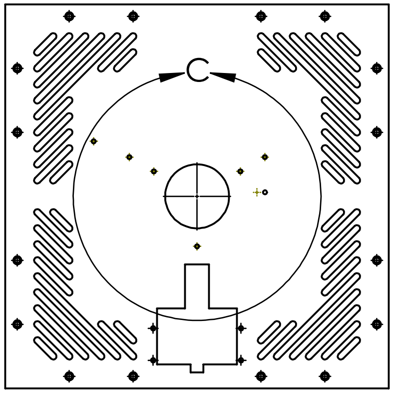

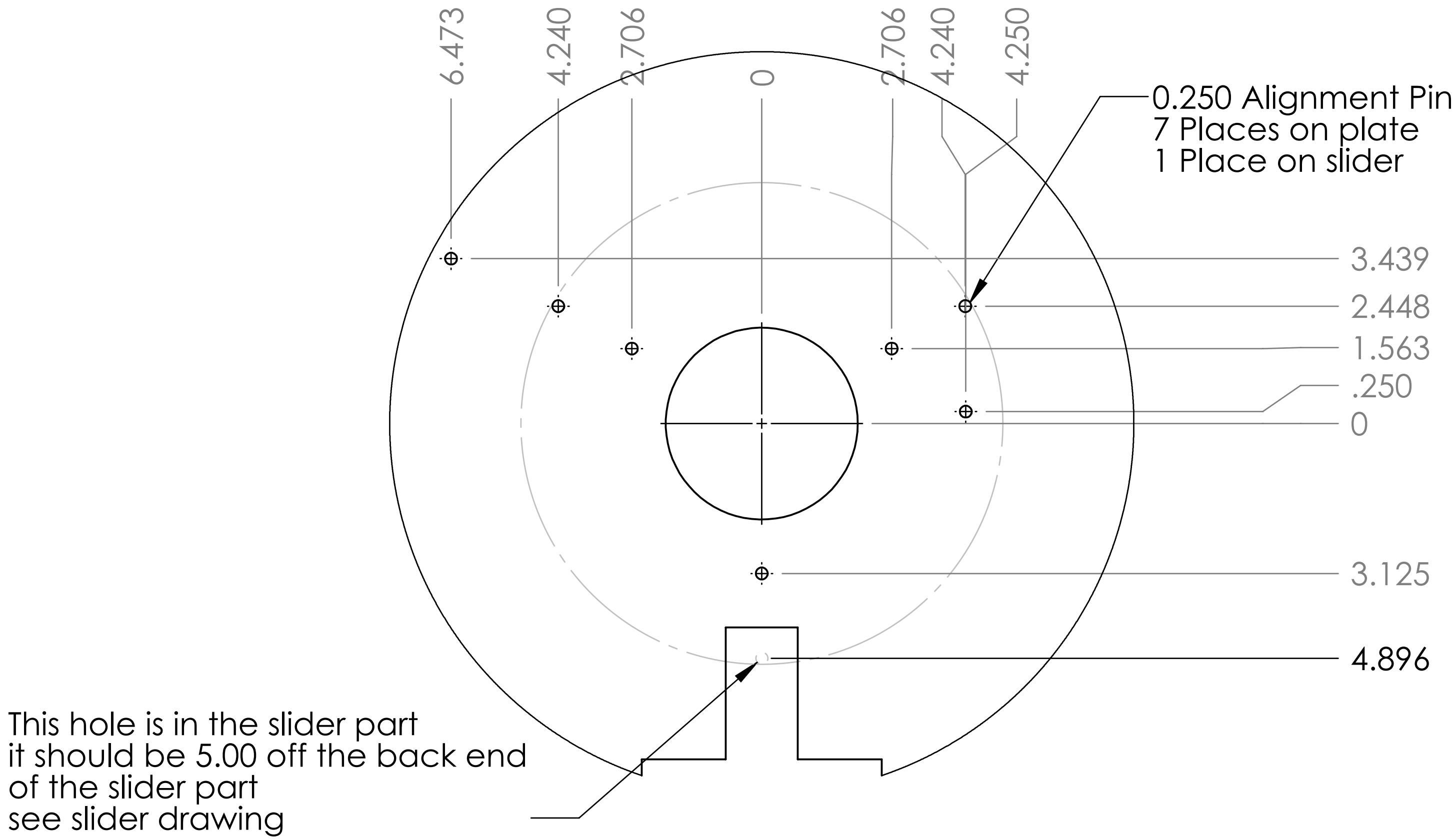

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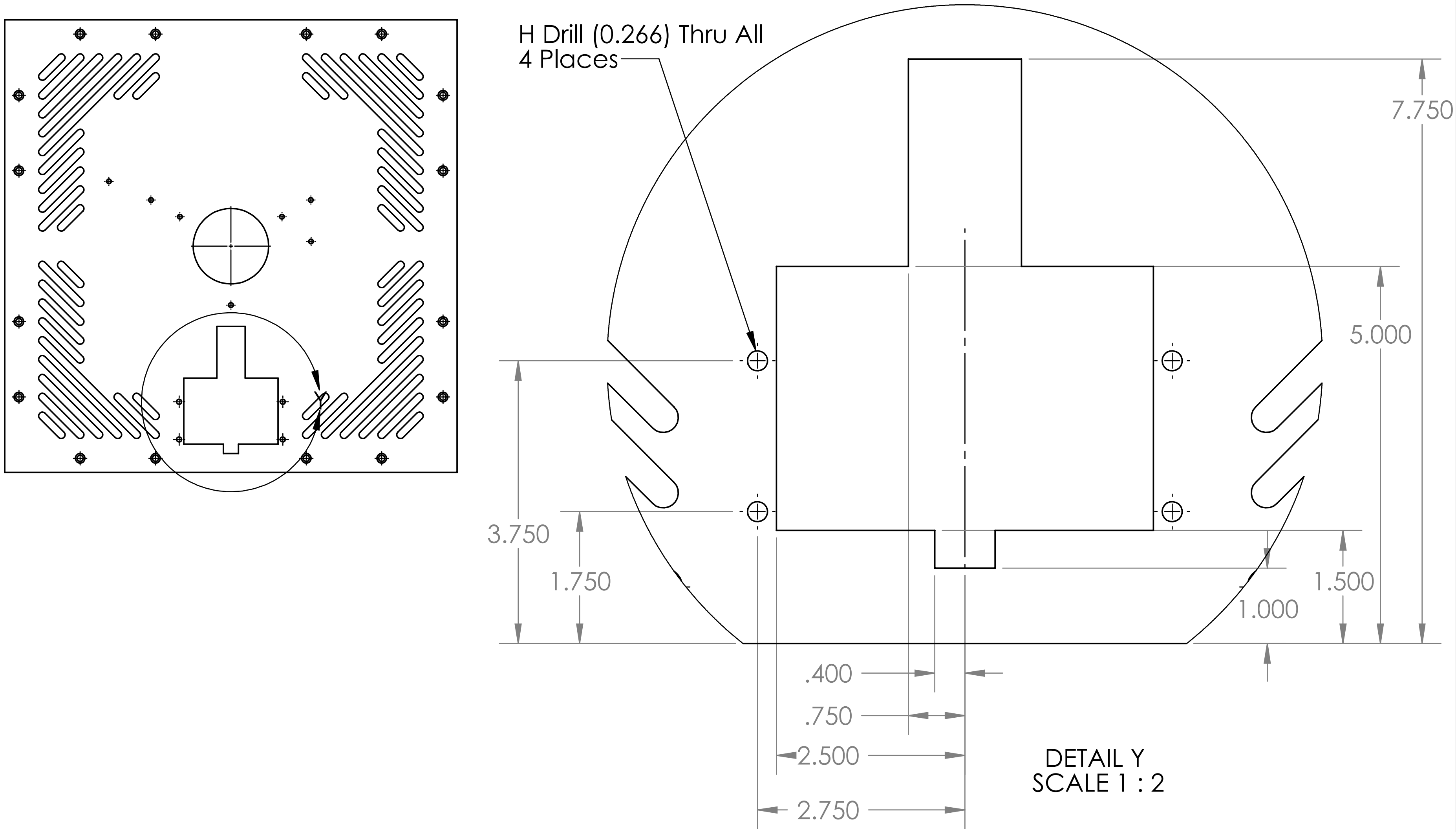

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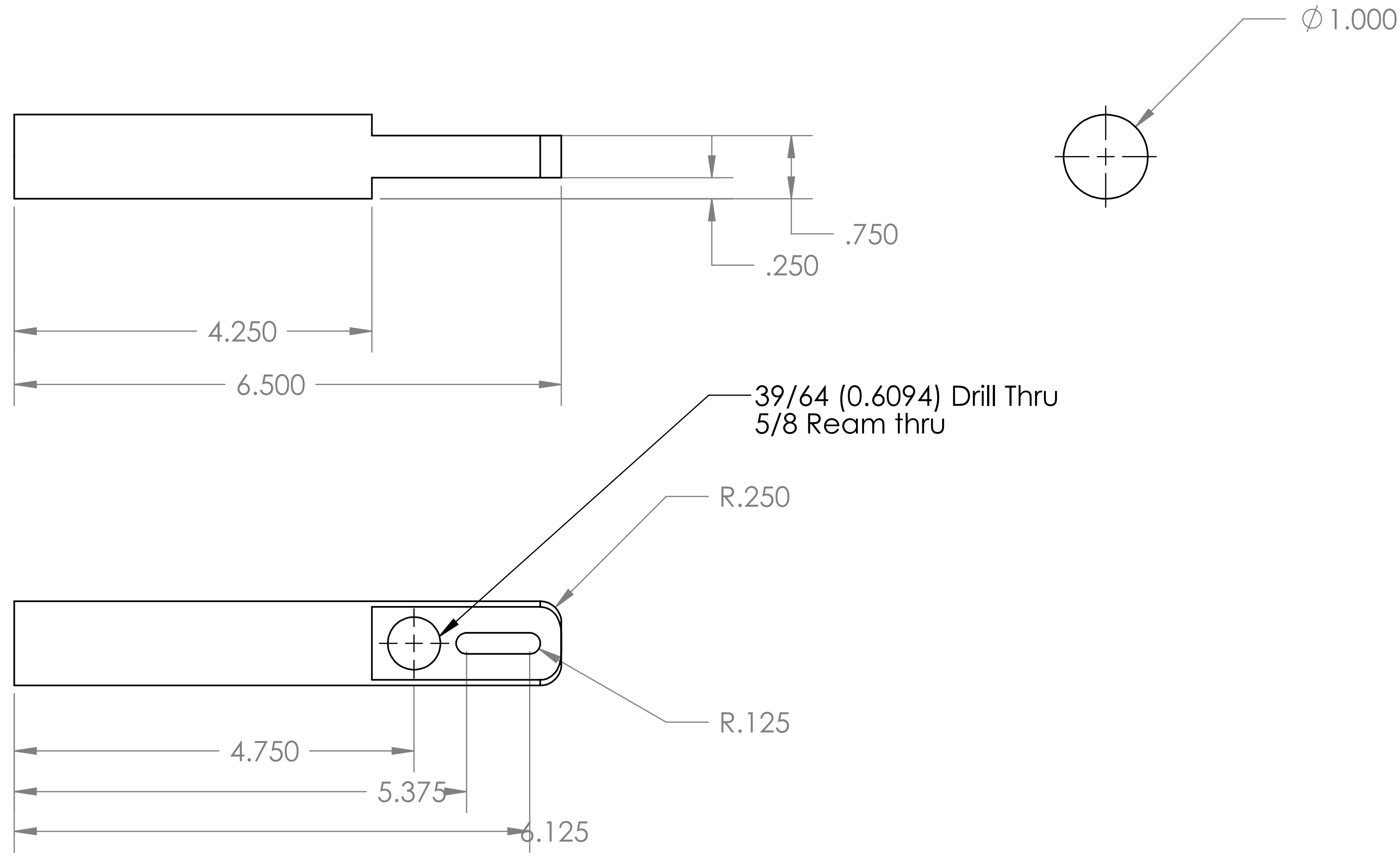

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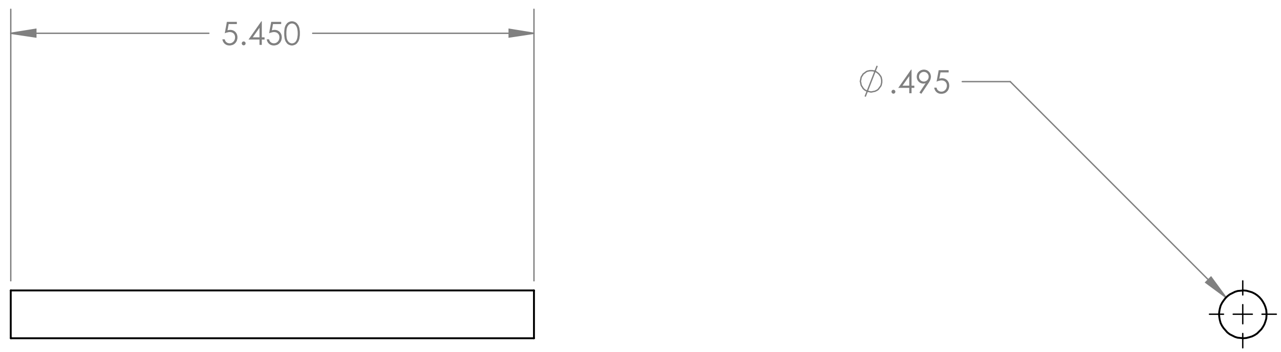

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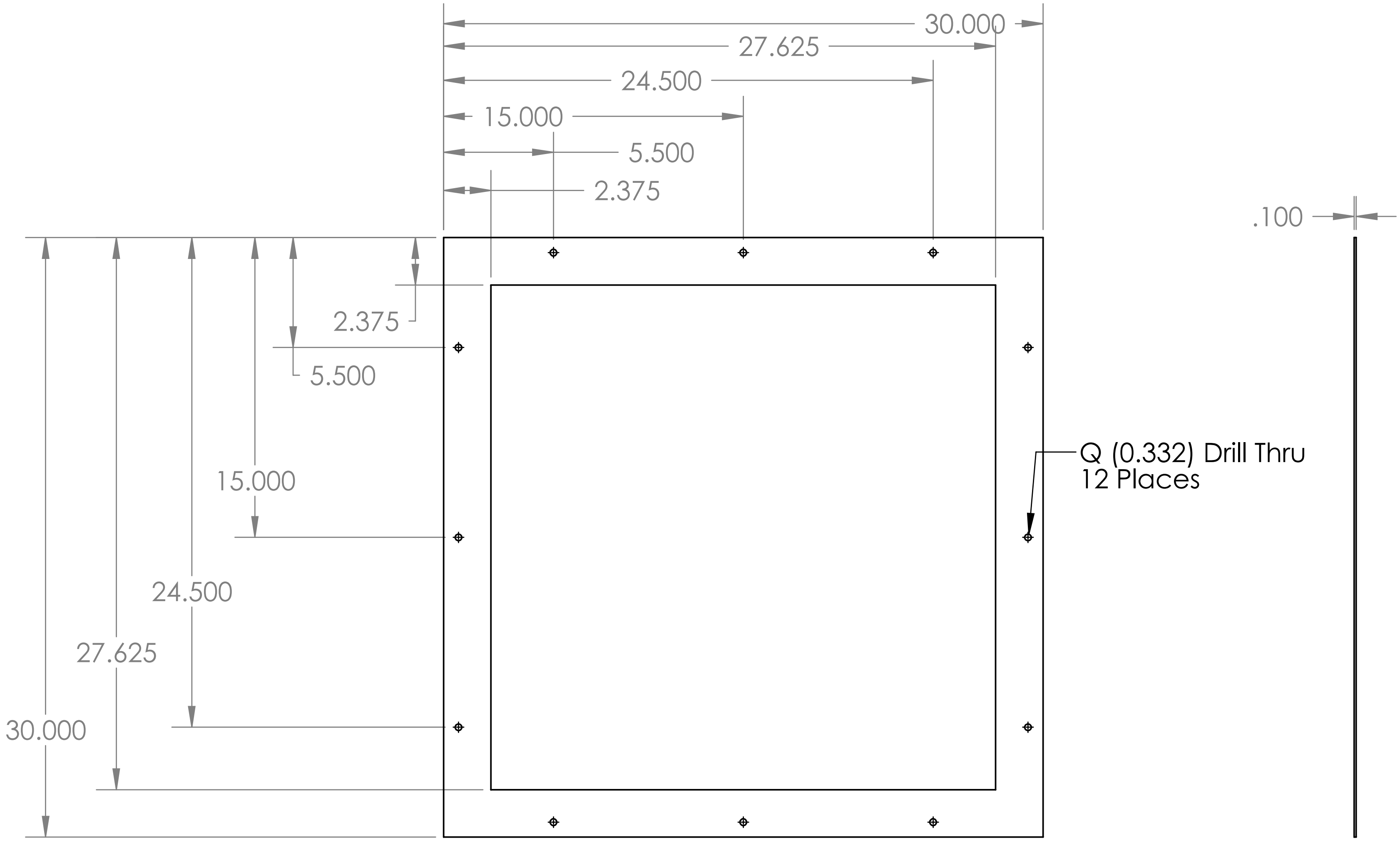
5 4 3 2 1



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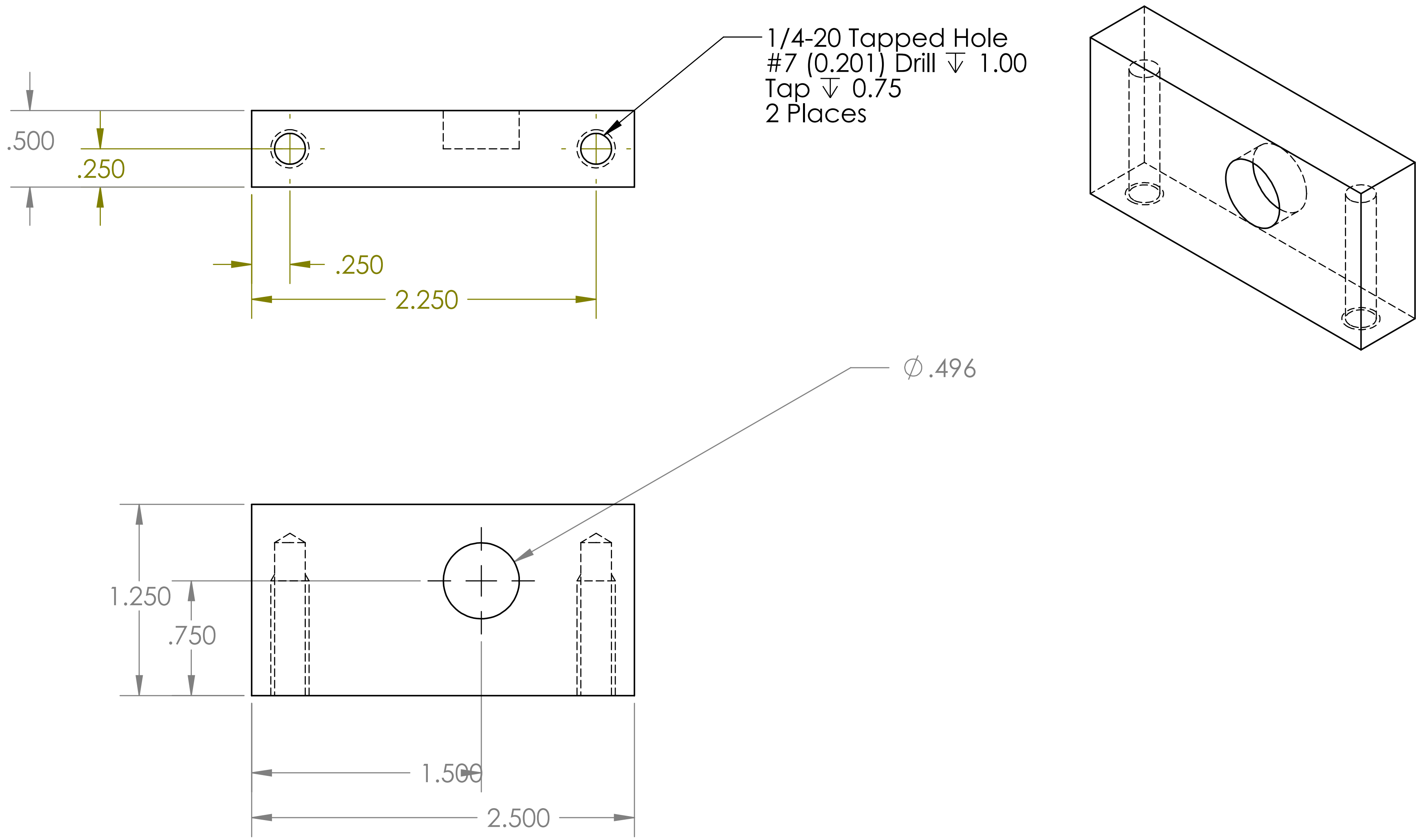

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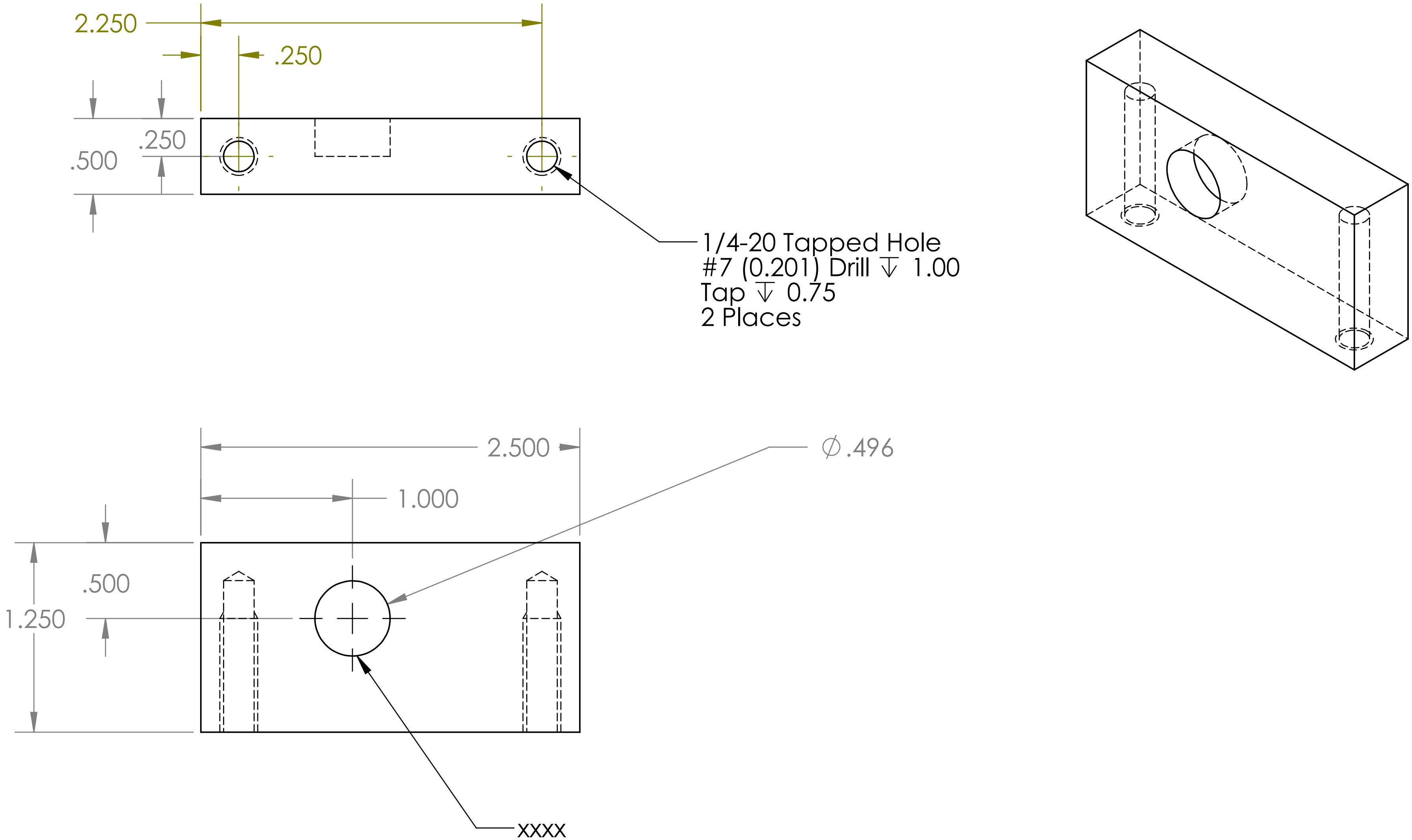

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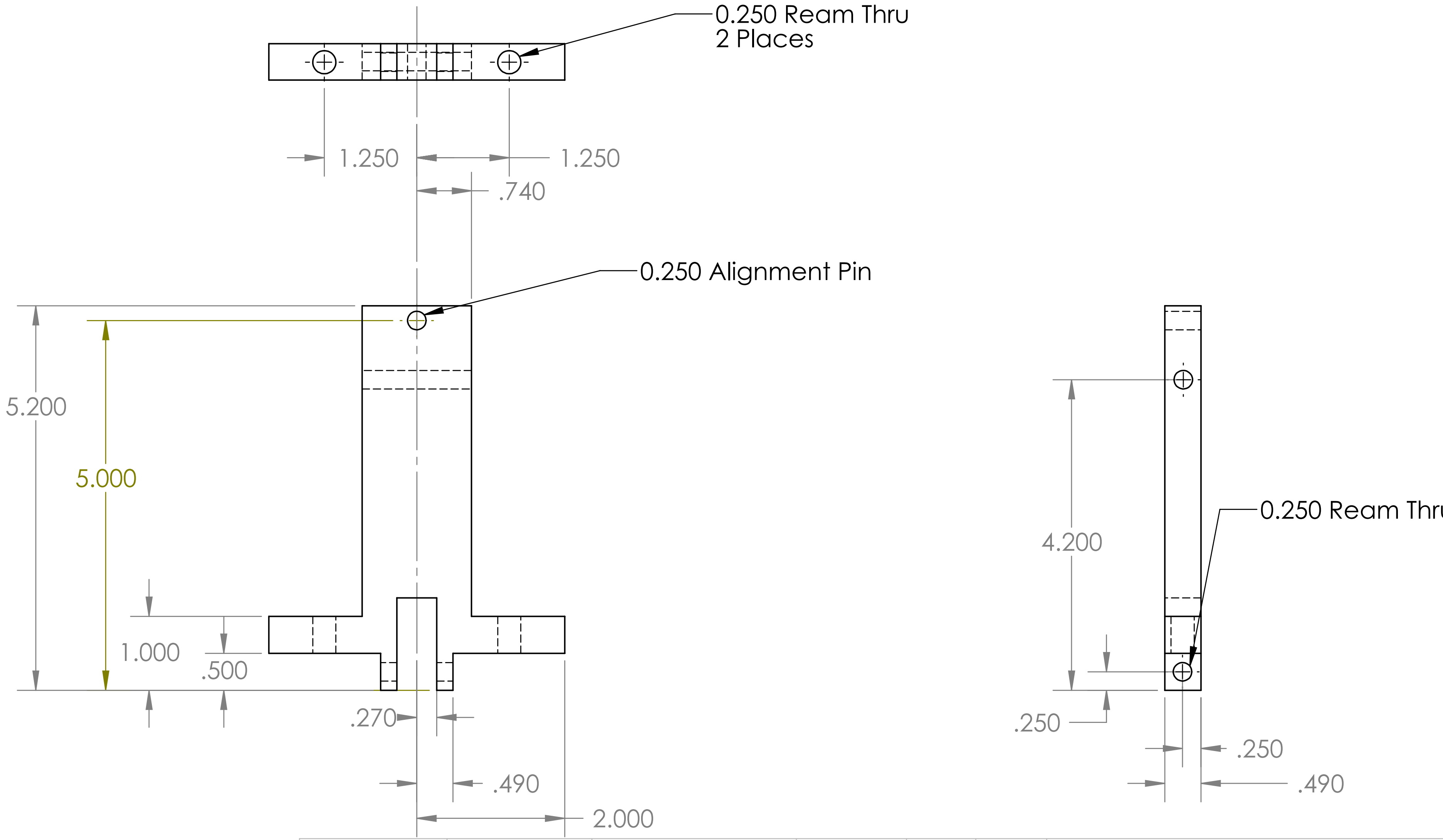

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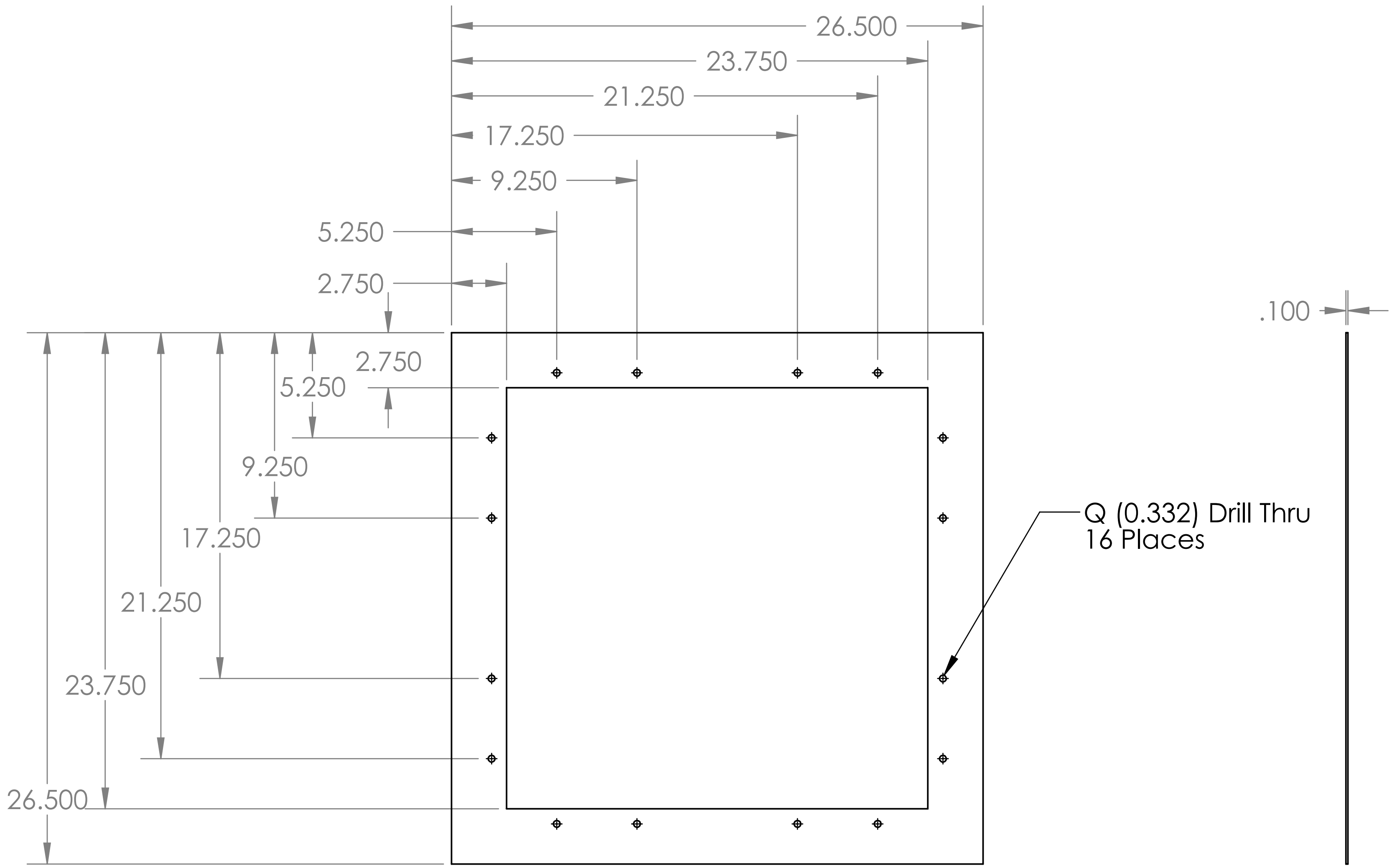

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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH

DO NOT SCALE DRAWING

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TITLE:

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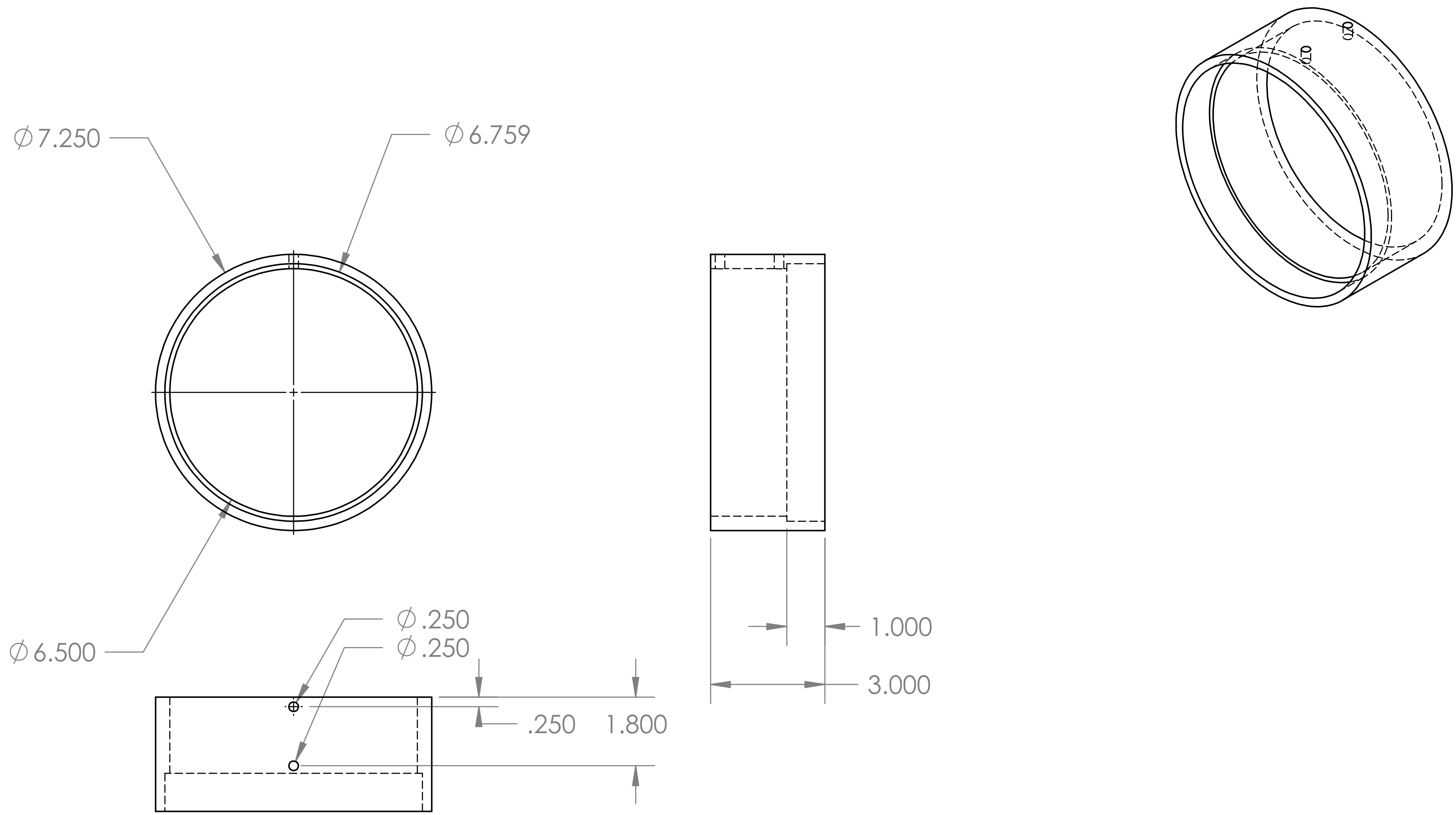
5 4 3 2 1



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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

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TITLE:

SIZE	DWG. NO.	REV
A	statoralignment	

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1